CROSSFLOW TURBINE DEVELOPMENTS & TESTING FOR ULTRA-LOW HEAD HYDRO APPLICATIONS

Report on Theoretical Modelling and Experimental Proving Work Undertaken with the Assistance of Energy, Mines & Resources – Canada

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EXECUTIVE SUMMARY

This shared-cost project has been concerned with assessing sloped blades as a means of improving the torque characteristics of the vertical axis Darrieus-type hydraulic turbine. The work was concerned less with efficiency than overall cost effectiveness, and the aim was to assess turbine configurations likely to lead to a smooth and reliable turbine.

The project was undertaken in two phases: the first both developed a mathematical modelling technique to project turbine performance with sloped blades, and also designed an experimental set-up; the second was primarily experimental, involving extensive running and measurement of a variable-geometry test turbine in N.R.C.'s flume tank in Ottawa. Theoretical and experimental results were later compared and conclusions drawn.

The first phase theoretical analysis indicated that a sloped bladed Darrieus type turbine would give a far smoother torque pattern (and hence less vibration) than a vertical bladed type, though at the expense of some power loss. Results were closely linked to the degree of slope. Experimental work in phase 2 showed that, although the trends of torque smoothing and power loss were apparent, the first phase projections of performance were not attained and were less than satisfactory.

There appear to be several reasons for this. Most significantly the sloped blades apparently introduce a third dimension to the streamflow which increases with blade slope angle, and which is negative on performance. In addition, with the straight blades used, greater slope gave a reduced turbine area so reducing possible net power.

There was significant disagreement between experiment and theory, much of which can be explained by the modelling program being unable to account for the third dimension of flow. Experimental data was used in an attempt to set the correct parameters for the theoretical model but it still did not prove possible to obtain good agreement between experiment and theory, even when modelling unsloped (vertical) blades. Questions remain about the validity of the computer model for predicting performance of turbines of the size and geometry used in these laboratory tests.

In summary, simply sloping the blades of a Darrieus turbine is

not the answer to reducing the torque pulse. If an effective means of suppressing (or making use of) the vertical component of flow imparted by the blades could be developed, it would greatly enhance the smoothness of this turbine type. This was an exploratory project which did not achieve the goals that had originally been expected, but which nevertheless added a number of new dimensions to the understanding of the Darrieus turbine and which will assist in its further development. It does not suggest that this line of approach taken should be dropped, though the insight gained from the work will focus futher development more effectively.

ACKNOWLEDGEMENTS

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1.0 INTRODUCTION

1.1 Background

As higher head hydro sites become more scarce and further from load centres, there has been an increasing interest in power from low head sites and from tidal and river currents. This has resulted in development work taking place throughout the world on devices which can efficiently convert this relatively diffuse energy resource into acceptably priced power or useful work.

Low head power struggles to be economically viable as it competes with more concentrated energy forms which are more readily converted into useful form. In short, a diffuse energy form requires larger energy converters per unit of useful energy produced than a concentrated one, and the equipment may therefore be relatively more expensive. However, the forces acting on the converters may be less, and savings in engineering sophistication can offset some of the additional material costs.

The Darrieus turbine (or derivatives based on the Darrieus concept) is one of a number of types of competing fluid energy converters possibly suitable for very low head applications. These have been listed extensively elsewhere (Ref. 1).

The vertical axis crossflow turbine (derived from the Darrieus turbine) has been under development for low head hydro applications for some ten years, and has some inherent advantages over other types of more established (axial flow) designs. Because of its basic simplicity it has the potential to be manufactured at costs far below conventional axial propellor designs. However, because of difficulties of installation, reliability problems and a consequent lack of effective site testing, its cost effectiveness and commercial viability remain to be proven. These factors, it should be noted, are not exclusive to the vertical axis Darrieus-type design.

1.2 Aim of work

This present project was designed to seek a solution to the reliability problem that has plagued this basic design, and which largely stems from the severe (and damaging) torque pulsing which has been prevalent in vertical axis crossflow turbine operation. It was also directed at determining what cost savings may be possible through design simplifications.

The objectives of this work were to:

1. investigate different blade mounting options and blade slope on turbine efficiency and torque characteristics which may prove possible on Darrieus and Darrieus-derivative crossflow turbines;

2. fabricate an experimental crossflow turbine with means of adjusting blade slope and angle of attack, and with variable means of blade mounting (end disks and struts) for performance testing in NRC's Hydraulics Laboratory.

The goal sought for a low head turbine is not necessarily high efficiency or low cost, but overall cost effectiveness and a delivered cost of power (or useful work) which competes successfully with other energy sources. A turbine which produces useful energy cost effectively as well as being simple, robust and easy to deploy seems likely to have a ready market in many small scale applications where conventional power sources are neither cheap nor readily available.

Progress towards the goals of robustness and cost effectiveness cannot be properly measured until developmental turbines are built and are operating in the field. The present project was a first step in this direction, aimed at a laboratory-based and theoretical analysis of design changes deemed likely to improve the vertical axis turbine. Questions remain as to what constitutes an "acceptable" performance in practice, and how the severity of the torque pulse may influence lifetime and reliability.

The work was undertaken in two phases. The first was concerned with developing a mathematical modelling approach to project the performance of different turbine configurations and also with designing a test bed turbine for second phase experiment work. As well as the experimental work (undertaken in the NRC's Ottawa laboratories), second phase work comprised further modelling in the light of the experimental work, analysis and interpretation of results.

1.3 Development rationale

In a previous study undertaken for the NRC (Ref. 2), Highquest identified that the severe torque pulse (or optimistically termed "torque ripple") associated with these vertical axis turbines may be alleviated by angling the turbine blades relative to the rotational axis of the turbine. The rationale for this approach is to broaden the swept angle (of azimuth) through which any single blade produces power, so spreading the delivered power over a wider arc of rotation than for a vertical blade.

As this project was designed to address the practical aspects of turbine cost as well as lifetime and reliability, we were aware of the need for a robust and easily constructed device. In this regard we felt that blades mounted on end discs (rather than support arms) would offer advantages in speed and ease of construction, as well as rigidity and, perhaps, less turbulence (and detrimental effect on performance) than separate support arms.

2.0 TEST-BED TURBINE DESIGN

2.1 Design Factors

The new test-bed was designed around an existing rig which has been installed in NRC's flume tank in Ottawa, and which thus constrained the basic turbine dimensions. Changes were necessary to modify the test rig as previously set-up to counter the shaft slop (which had resulted in difficulty in properly measuring torque) as well as to allow variations in blade positioning.

The redesigned set-up improved the reliability of experimental results, as well as allowing a variable geometry test turbine, where blades may be sloped to different angles.

The basic dimensions of the test turbine remained as before:

Rotor diameter	630mm
Blade height (vertical blades)	640mm

giving a swept area of .4032 m^2 . Sloped blades have the effect of "necking" the turbine and reducing the overall frontal area slightly.

This sloped bladed approach could use both helical or straight blades. Their relative advantages are discussed below.

Blades:

The revised rig is designed to operate with NACA 0015 3" chord blades and with adjustable (taper lock) bushings on the shaft/support arm connection. This blade section, while probably not giving best performance in a small test turbine, was chosen because of its ready availability.

As blades are sloped relative to the axis of water flow, they become twisted along their longitudinal axis to retain a constant angle of attack. To maintain a constant radius (and hence a constant turbine area relative to flow) blades would be helical.

Helical blades have this significant advantage of a constant turbine area irrespective of blade slope, but blade stresses are higher than for straight blades. The lower stresses on straight blades must be weighed against the loss of turbine area (and consequent loss of power output). Straight blades have the important advantage of ease of fabrication, as there are practical difficulties in forming a helical blade. This latter factor was overruling in the case of the present work, and all experiments were conducted using straight blades. These were twisted by a lever in the workshop prior to installation on the

test turbine.

Straight blades also correlated with the stress analysis which was calculated on a straight, longitudinally twisted blade.

Shaft:

The initial intention of removing the central shaft and transmitting the torque loads through the blades to end disks was unfortunately determined not to be possible in this program for two reasons:

a) the existing bottom bearing arrangement gives insufficient rigidity to the rig with the central shaft removed. A new bottom bearing arrangement would be required, entailing significant structural alterations to the flume tank and costing more than the project could support;

b) the blade/support arm connections (two machine screws) would shear under the starting torques.

This could have been overcome by leaving the shaft in place, and the wake effect of a small diameter shaft, such as fitted to this test rig, was determined not to affect downstream blades significantly.

A new shaft was designed with taperlock fittings which allow the hubs to be moved for the various blade positions. Two sets of hubs were mounted on the shaft throughout the tests: one accomodated the 1, 2 and 4 bladed configurations and the other the three bladed.

Blade supports:

In designs with a central shaft and support arms it is the shaft which transmits much of the torque. The argument for removing the central portion of the shaft and transmitting torque through the blades and blade supports to the power take-off is that it would reduce downstream flow turbulence from the shaft and support arms, and would likely add to efficiency. The blades would then be mounted on circular end plates (or disks), which are stronger and more rigid than support arms.

Theoretical analysis of the relative drags of support arms and end disks shows that disks have a higher drag effect, conversely however, they offer a benefit of confining the vertical component of flow induced by sloped blades.

Unfortunately, the difficulties and time involved in aligning the shaft of the test rig strongly discouraged using disks for

practical reasons of setting up. Because of this, and also because the central part of the shaft was not being removed, it was decided to use the support arms that were available from previous experiments.

2.2 Turbine geometry

The turbine geometry is more complex than initially apparent, and particular care was necessary in dimensioning the blades for machining.

The blades, although sloped at an angle to the vertical were better defined by azimuth angle between top and bottom support arms (nominally at the ends of the blades). This angle could be better used to defined geometry with turbines of various aspect ratios. Fig. 2.1 shows the geometrical set-up for assessing turbine dimensions.

2.3 Design

The revised test bed turbine design is shown in Appendix A. A new shaft and hubs were fabricated but the support arms were those used in previous tests. A flywheel was attached between the torque transducer and the generator to assist in smoothing fluctuations in speed (an approach which holds merit in production turbines and could be beneficially applied).

Blades were fabricated from 6063 T54 aluminum, and are of the same section and stock as those used by Nova Energy Ltd. in their TOR5 turbine. Before being delivered to NRC, Nova undertook some trial twisting of the blades, managing a 30 degree twist over 1 m length with 115 Nm of torque, and so determined they could be suitably mounted in the test rig.

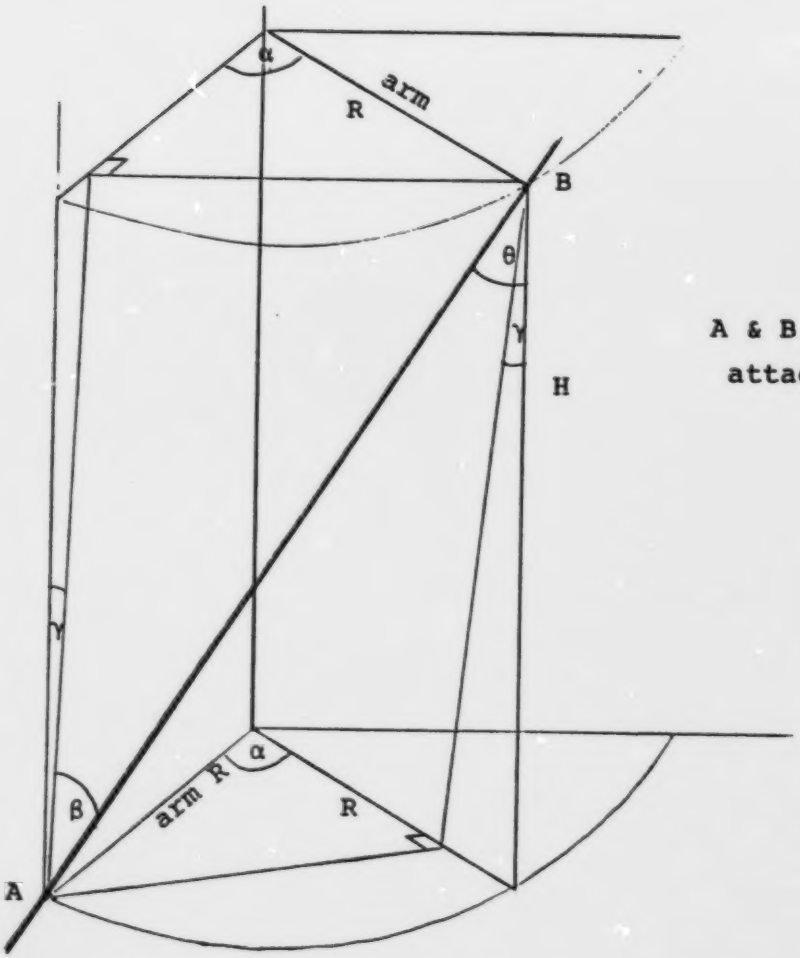
2.4 Blade preparation

Sets of blades were prepared for five different turbine geometries:

azimuth angle	=	0°	(blades straight and vertical)
	=	15°	
	=	30°	
	=	50°	
	=	80°	

Appendix B shows blade machining and drilling requirements, denoting the angles of twist and all necessary dimensions. Because of the three dimensional nature of the test turbine this proved to be a complex task, which was tackled by Thierry Faure of NRC who developed a suitable methodology.

The extruded blade sections were twisted to the required angle by hand in a rig prepared by N.R.C., and the slots and countersunk screw holes machined after twisting.



α = azimuth angle (= 0^{o}, 15^{o}, 30^{o}, 50^{o} & 80^{o})

β = angle to machine flats on the blade

FIGURE 2.1

TURBINE GEOMETRY

3. TURBINE EXPERIMENTS

3.1 Purpose and organization

The purpose of the experimental work was to validate the theoretical performance modelling and to measure the sensitivity of both power output and torque to variations in blade slope.

This phase of work was carried out under Highquest's direction by NRC staff using the flume tank at NRC's Hydraulics Laboratory in Ottawa. This facility has been used for all the laboratory-based vertical axis hydro turbine experiments carried out to date in Canada.

The experimental set-up in the flume tank required certain changes to incorporate the new test turbine, and to improve the quality and consistency of readings from the torque transducer, (see Section 2). The test-turbine assembly needed time and care to set it up as three pieces connected by rigid couplings had to be aligned. Figure 3.1 shows the new experimental rig in the flume tank, and appendix A includes a general arrangement of the test set-up.

The dimensions of the flume tank determined the test turbine radius and blade length and hence fixed the turbine's aspect ratio. With these dimensions, previous work suggests that a narrower blade section than the 75mm section used in this series would be more efficient. However, efficiency was not the most important aspect of these tests, and with the ready availability of the 75mm section it was felt to be quite suitable for the purposes of this work.

3.2 Control and data measurement

Five channels of data were recorded by a data logger:

rotational position
r.p.m.
water level upstream
water level downstream
torque

Torque signal sensitivity is 22.66 Nm/volt. Rotational position and torque were transmitted to a chart recorder. Typical output is shown in Figure 3.2.

All data was transmitted to the hydraulic laboratory's computer for processing. This is discussed in Section 3.5 following.

Measurements of power and r.p.m. were made from the torque transducer; flow rate through the turbine was determined by measuring the height of the water both up and downstream of the turbine (the flume tank being calibrated for this). The velocity through the rotor was calculated from flow rate, and tip speed ratio determined from the r.p.m. and velocity. These calculations were carried out automatically by the computer.

A rotation sensor was fitted to the shaft to give a signal to the chart recorder each revolution. This was used to determine the torque relative to the blade positions.

Rotational speed was set by the generator configuration, and was varied through the range of 35 - 140 r.p.m. (higher tip speed ratios being more appropriate for lower solidity turbine configurations). Water velocity through the turbine was controlled by a variable pitch pump adjacent to the flume tank.

3.3 Experimental procedure

Following the zeroing of the torque transducer, a series of calibration runs were initially completed to determine bearing losses and the drag factors of the support arms and blades.

Firstly, the turbine was motored in air (hence with negligible hydraulic drag) to measure bearing losses. It was then motored at various r.p.m.'s in still water in a one blade configuration (2 arms) both with and without a blade to measure both the arm drag factor and, by deduction, the blade drag factor. (This was later repeated in a three blade configuration, i.e. 6 arms). It was later motored with support arms but without blades in a stream velocity of 0.84 m/sec.

Plots of torque against rotational speed in Fig 3.3 show that the bearing losses decreased over the course of the experimental work as the lubrication characteristics changed and the bearings wore in.

During the course of these runs, a 15 - 20 hertz resonance was apparent from the torque transducer output, which was attributed to the mechanical vibration of the whole structure. This was prevalent throughout the entire experimental program. When data was processed, this signal was eliminated by filtering all frequency components above 7 hertz.

Once the test turbine was prepared in its desired configuration it was motored at a set speed and the circulation pumps activated to fill the flume. When steady state was reached (determined by

up and downstream water level sensors properly filled) sample output was switched from the data logger to the computer for a period of 20 seconds. At the same time direct readings of torque for two to three revolutions of the turbine were obtained from the chart recorder.

3.4 Experimental runs

Following the calibration runs a total of 237 experimental runs were completed. Variables were:

Number of blades
0, 1, 2, 3, 4
Pitch angle of blades (angle of attack)
0, 2, 4, 6, 8
Slope angle of blades (relative angle of support arms)
For 1, 2 & 4 blades: 0, 8.2, 15.9, 24.9, 35.2
" 3 blades: 0, 9.8, 19.2, 25.1, 35.5 *
Rotational speed
stepped over the range 35 - 140

* the slope angles are different for three bladed turbine because of the position of the test rig support arms. (See Fig. 3.1)

Because of the large number of experimental runs possible, a set of runs was undertaken which gave representative results for the turbine operation in a range of configurations and operating conditions.

Actual runs completed and processed are listed on Tables 3.1 to 3.5.

Typical raw and filtered output from the torque transducer is shown in Figure 3.4.

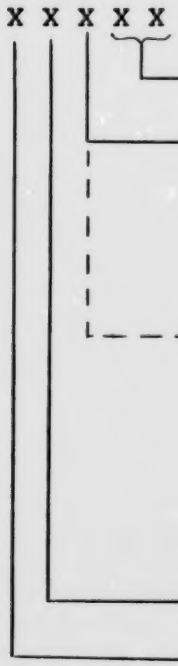
3.5 Data Processing

The raw data from the torque transducer was filtered to remove the signal resulting from the resonance of the structure. A 7 hertz low pass filter effectively removed these 15 - 20 hertz signals. Sampling rate was 100/sec.

Each test run was sampled by the computer for a period of 20 seconds, and the torque data subjected to fourier analysis after filtering. Typical output from the computer analysis is shown in Figure 3.5.

This 20 second sampling period addressed certain of the experimental errors relating to the fabrication of the test equipment. The raw torque output showed variations between blades of the "same" geometry and type, pointing to small differencies in manufacture. This demonstrated the sensitivity of the output primarily to small changes in angle of attack, but it was not possible to quantify these due to the difficulties in determining the actual blade dimensions and geometries as installed. Data results were averaged over this sampling period.

Experimental runs were coded as follows:



r.p.m.: for r.p.m. > 100, use last two digits only

test series:

H = 2° blade pitch angle
I = 6° "
J = 4° "
K = 8° "
L = 0° "

calibration series:

M = zero water velocity - with blades - 2° pitch angle

N = zero water velocity - no blades
P = ~0.84 ms^{-1} water velocity - no blades.

Number of blades - 2 arms per blade

Blade slope angle (measured to vertical)

For 1,2 and 4 blades:
A = 0° (vertical blades)
E = 8.2° ; arm angle = 15°
D = 15.9° ; " = 30°
B = 24.9° ; " = 50°
C = 35.2° ; " = 80°

For 3 blades:
F = 9.8° ; arm angle = 15°
G = 19.2° ; " = 30°
H = 25.1° ; " = 41.4°
I = 35.5° ; " = 65.1°

3.6 Discussion of results

3.6.1 Bearing losses

The initial calibration of bearing losses showed that the torque necessary to turn the shaft was about 1 Nm irrespective of the rotational speed. This did not, however, remain constant over the course of the experiments as the lubrication characteristics changed and the bearings wore in.

Bearing losses were measured several times over the course of the experimental work and the computer program adjusted accordingly.

3.6.2 Arm and blade losses

Fig 3.6 shows plots of torque against rotational speed for calibration runs in still water. Analysis of these data shows power losses of an isolated support arm and of an isolated blade in Fig 3.7. Note that the power loss increases only slightly when the water is moving (i.e. in working turbine mode).

3.6.3 Experimental runs

1. Effect of pitch angle

Most experimental runs were undertaken with blade pitch at 2° , (i.e. a 2° toe-out of the leading edge of the blade). In two cases (vertical, 3 & 4 blade configurations) pitch angles of 2, 4, & 6° were compared and in one case (2 bladed with blade slope of 35.2°) 0, 2, 4, 6 & 8° pitch angles were tried.

Curves of power coefficient against rotational speed are shown for these 3 sets of runs, indicating an optimum pitch of between 5 & 6° . Note that r.p.m. is plotted rather than tip speed ratio because tip speed ratio is not constant for a variable radius (straight sloped bladed) turbine.

Our experiments were therefore mainly undertaken with a suboptimal pitch angle for maximum power output (as well as a suboptimal blade section). This was not a major concern for the purposes of this work however, which concentrated more on relative torque variations and net power.

Pitch angle had a small, though not significant, effect on peak to peak torque levels over the speed range tested (Figs. 3.8 to 3.10). Note from Figures 3.9 and 3.10 that the three bladed turbine appeared to have markedly more torque variation than a four bladed machine.

2. Effect of number and slope of blade on torque characteristics.

Figures 3.11 to 3.15 show standard deviation of torque against rotational speed for 1, 2, 3 & 4 bladed turbines. A 4 bladed machine appears to present a consistently smoother torque pattern than other blades over optimal working range.

Counterintuitively the 4 bladed turbine is consistently and significantly smoother than a 3 bladed over the entire range.

Figure 3.16 shows comparison of 4 bladed machines (standard deviation vs r.p.m.) for different slopes, showing that the torque pulse reduces significantly with increasing blade slope.

Reduced torque pulse is however gained at the expense of lower net torque (and often reduced power, depending on rotational speed). This is covered below.

Valid concern must be expressed about negative torque pulses which are apparent in many of the experimental runs undertaken, and which do not augur well for long life and reliability. However, the poor power performance found in these experiments is not surprising in view of the acknowledged downplaying of optimization for efficiency and power. If these aspects were addressed, we feel that torque would be unlikely to go negative.

3. Effect of number and slope of blades on net average torque

Figures 3.17 to 3.21 show average torque against rotational speed in r.p.m. for blades of different blades and slopes, and now introduces the hitherto neglected aspect of turbine energy performance. The highest torque is produced with vertical blades, with average torque (and also power output) declining as blade slope increases, clearly shown on Fig. 3.22.

Examination of Figures 3.17 to 3.19 reveals that for the smaller angles of blade slope two and four bladed machines give better torque than either one or three blades. This is not so pronounced at greater angles of slope, and Figure 3.20 shows that a three bladed turbine is actually better with a 25° blade slope. This improvement, however, is not so marked for the 35° slope (Fig.3.21).

4. Effect of number and slope of blades on torque pulse / power performance.

Figure 3.23 shows plots of $\frac{\text{Net average torque}}{\text{Torque std. deviation}}$ vs r.p.m. for a four bladed turbine with different degrees of blade slope. This indicates relative performance embracing both torque pulse and power factors, with higher values indicating "better" overall performance. The curves show that the vertical bladed machine appears to give the best overall performance when judged by this uncalibrated means of comparison, excepting what appears to be a rogue reading at about 45 r.p.m. where maximum blade slope appears much better than the other configurations.

However, the better performance of the vertical bladed machine is not overwhelming.

A more detailed assessment of the performance of different turbine configurations is undertaken in Section 5 following, in which both theoretical and experimental results are reviewed, compared and analysed.

TABLE 3.1

BLADE SLOPE: 0° SUPPORT ARM ANGLE: 0°

NUMBER OF BLADES	PITCH ANGLE (Degrees)					ROTATIONAL SPEED																						
	0	2	4	6	8	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140
1		X							X			X		X		X		X		X		X		X		X		X
2		X								X			X		X	X	X	X		X		X						
				X						X			X			X	X	X		X		X						
3		X						X		X		X		X		X		X										
			X					X		X				X		X												
				X				X			X		X	X	X		X		X									
4		X						X		X		X	X	X	X		X											
			X						X		X	X	X	X	X		X											
				X					X		X	X	X	X	X		X											

TABLE 3.2

BLADE SLOPE: 8.2° SUPPORT ARM ANGLE: 15°

NUMBER OF BLADES	PITCH ANGLE (Degrees)					ROTATIONAL SPEED																						
	0	2	4	6	8	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140
1		X									X			X		X		X		X		X			X		X	
2		X					X		X			X		X	X	X		X			X		X					
4		X						X		X		X	X	X		X		X										
Blade slope = 9.8° Support arm angle = 15°																												
3		X						X			X		X	X	X		X		X									

TABLE 3.3

BLADE SLOPE: 15.9° SUPPORT ARM ANGLE: 30°

NUMBER OF BLADES	PITCH ANGLE (Degrees)					ROTATIONAL SPEED																						
	0	2	4	6	8	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140
1		X									X			X		X		X		X		X			X		X	
2		X								X			X		X	X		X			X							
4		X							X		X	X	X	X	X	X		X										
Blade Slope: 19.2° Support arm angle 30°																												
3		X							X		X		X	X	X		X		X									
			X						X		X		X	X	X		X		X									

TABLE 3.4

BLADE SLOPE: 24.9° SUPPORT ARM ANGLE: 50°

NUMBER OF BLADES	PITCH ANGLE (Degrees)					ROTATIONAL SPEED																				
	0	2	4	6	8	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140
1		X									X			X		X		X				X			X		X	
2		X							X				X		X	X		X			X							
4		X				X			X		X	X	X	X	X		X											
Blade slope: 25.1° Support arm angle: 41.4°																												
3		X						X			X		X	X	X		X		X									
				X				X			X	X	X	X	X		X		X									

TABLE 3.5

BLADE SLOPE: 35.2° SUPPORT ARM ANGLE: 80°

NUMBER OF BLADES	PITCH ANGLE (Degrees)					ROTATIONAL SPEED																						
	0	2	4	6	8	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140
1		X									X			X		X		X		X		X			X		X	
2	X										X			X		X		X		X		X						
		X					X			X			X		X		X		X									
			X									X			X		X	X		X			X					
				X							X			X			X		X		X		X					
					X						X			X		X		X		X		X		X				
4		X				X			X		X	X	X	X	X		X											
Blade slope = 35.5° Support arm angle = 65.1°																												
3		X						X			X	X	X	X	X			X										
			X					X			X	X	X	X	X	X			X									

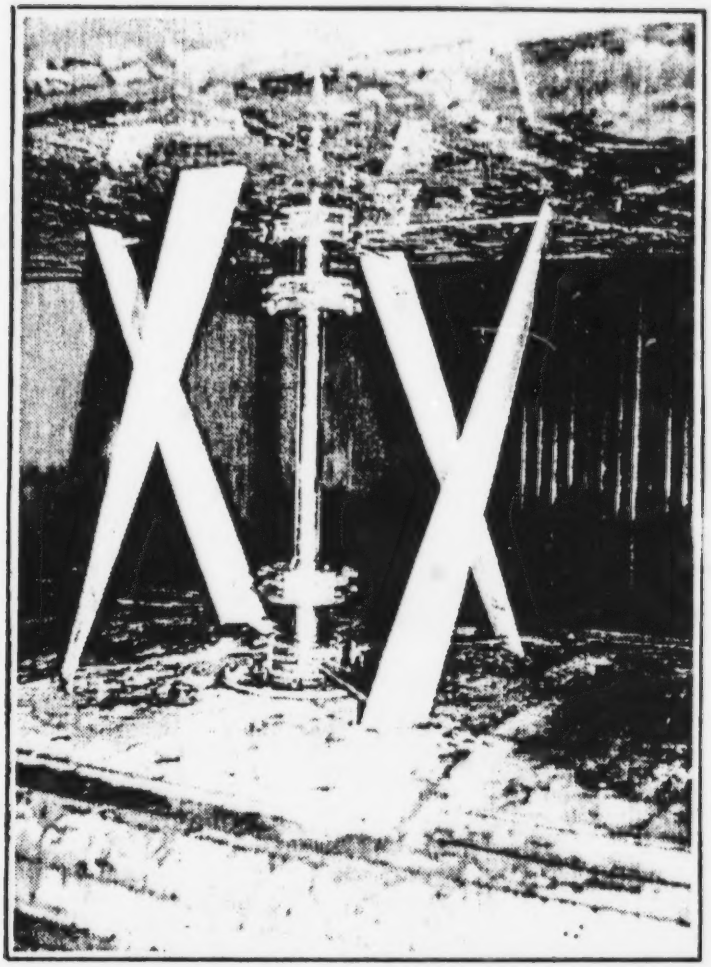


FIGURE 3.1 Test turbine in NRC flume tank (25^{o} blade slope)

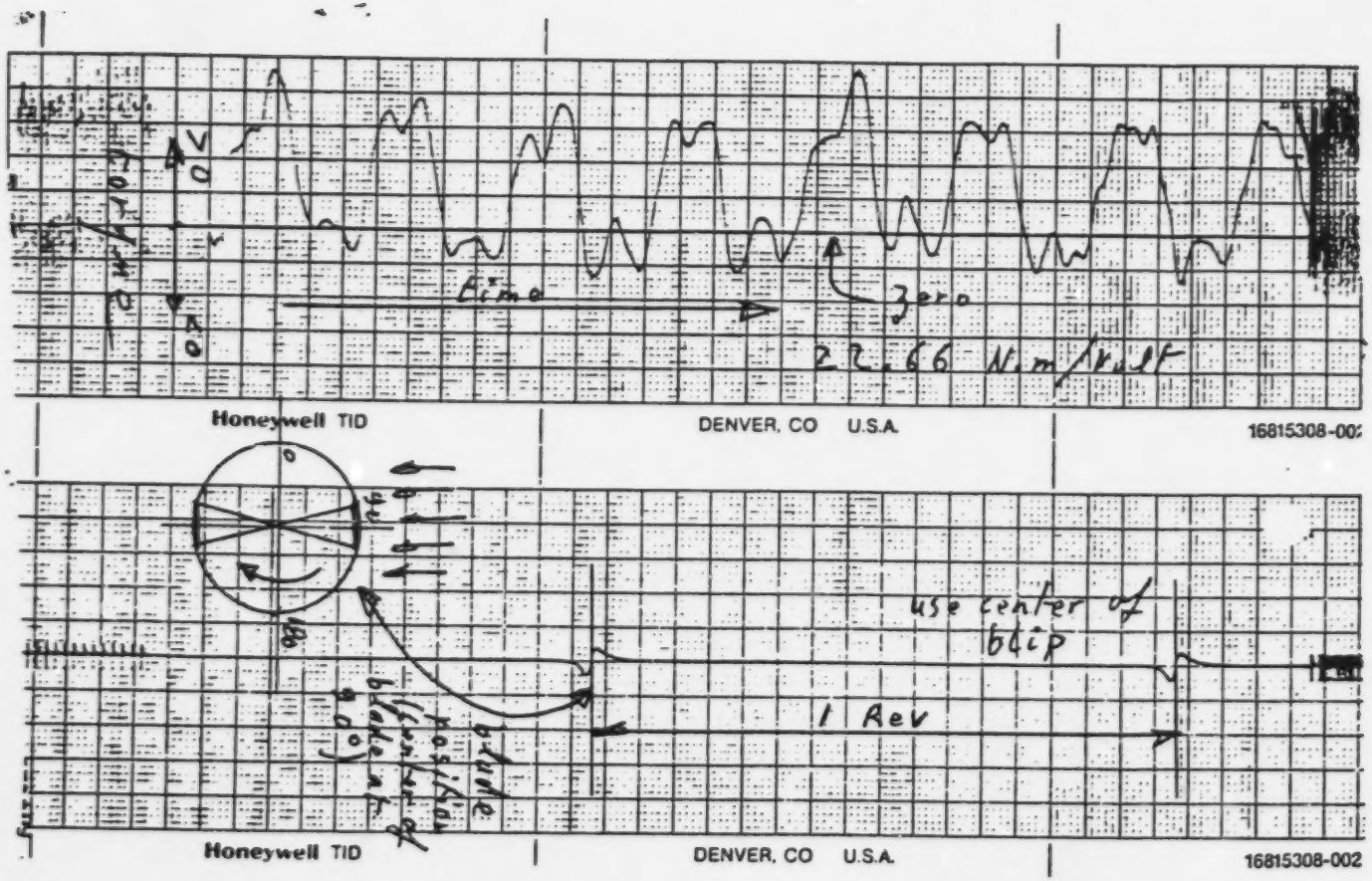


FIGURE 3.2 Chart recorder output

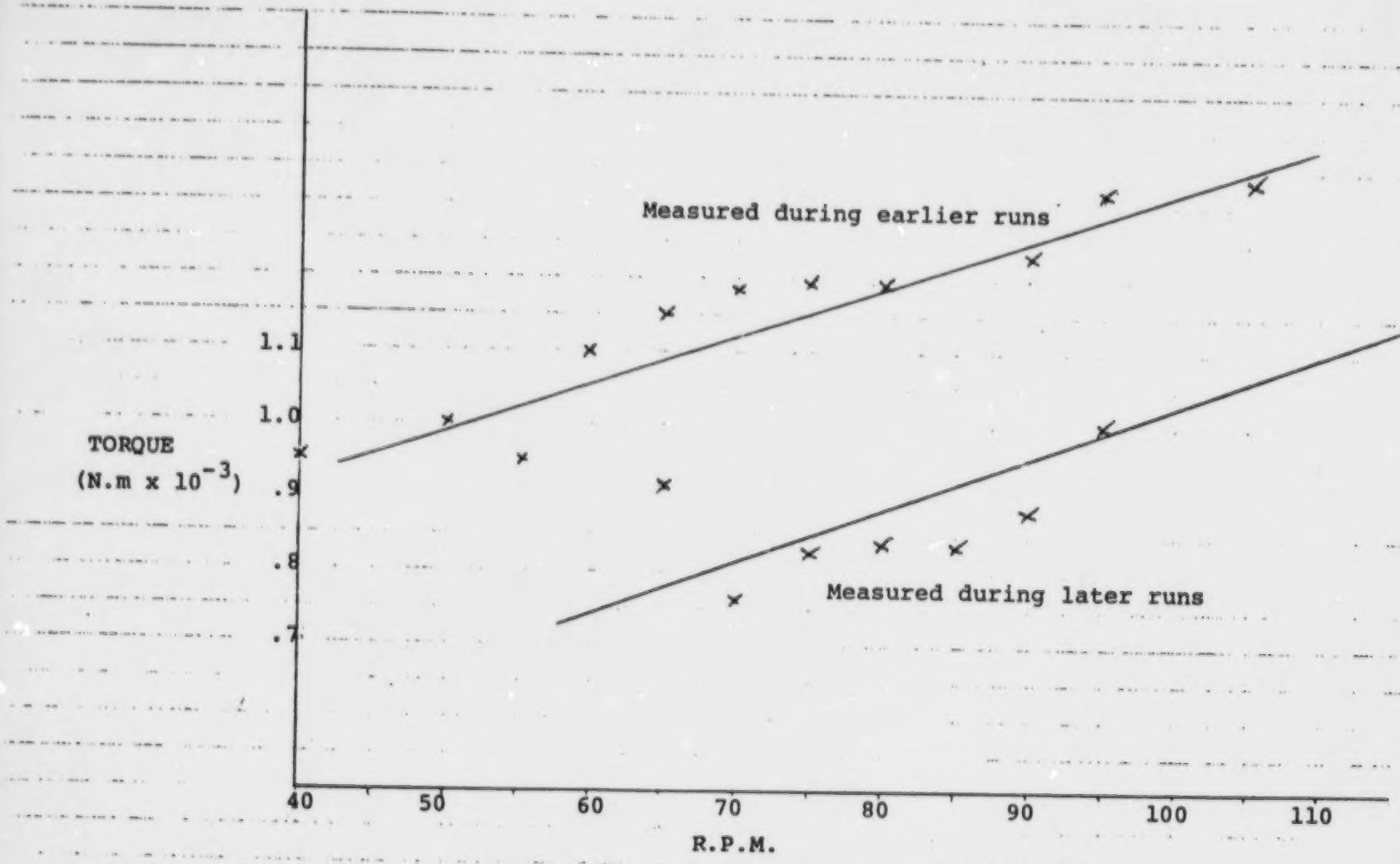


FIGURE 3.3 - TORQUE DUE TO BEARINGS

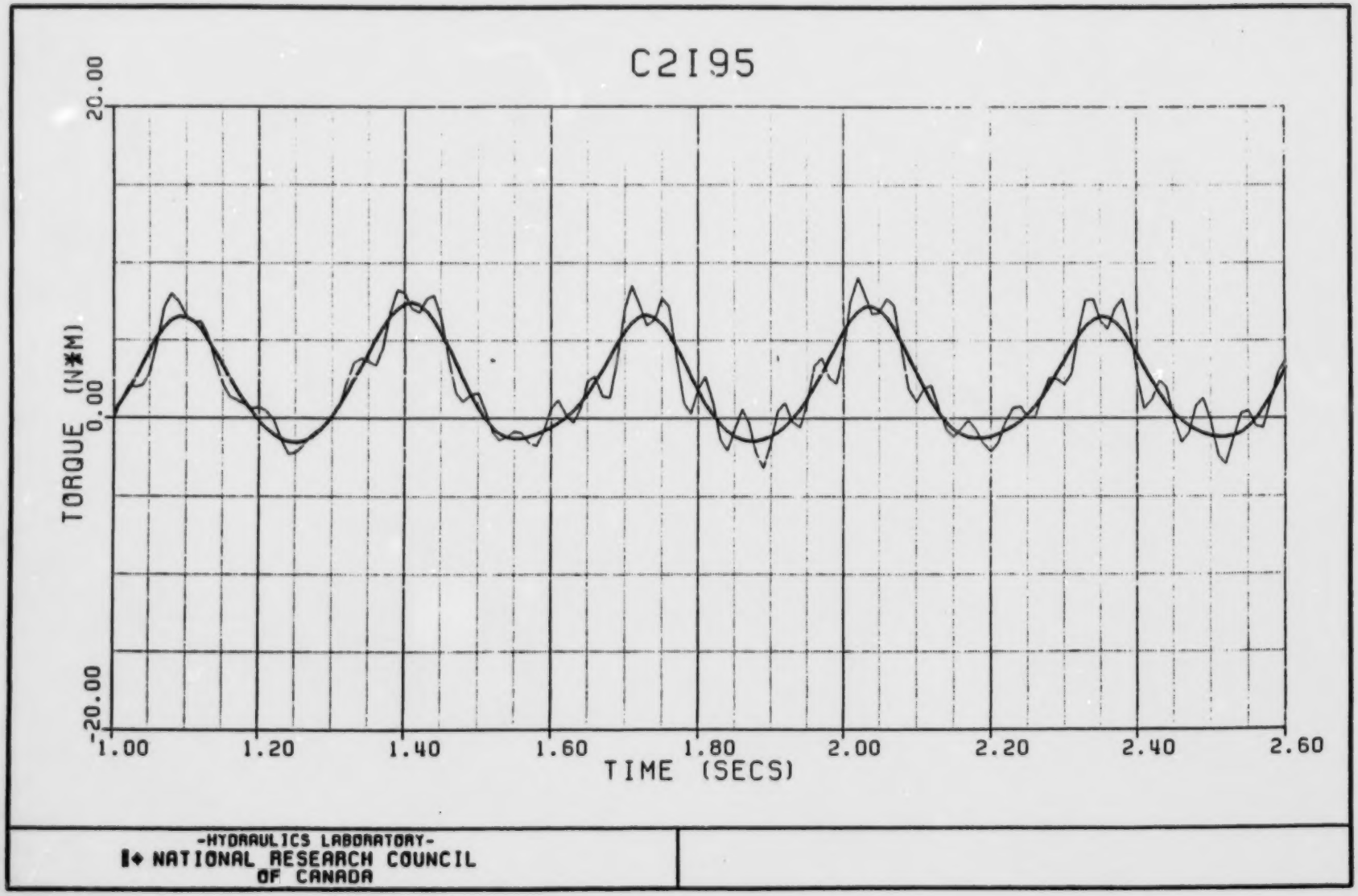


FIGURE 3.4 Example of raw and filtered output

VAHT - SLANTED BLADE TEST (HYQUEST) - FREETREAM - NO DUCT

```
EXPERIMENT ID:B4H30        FRONTAL AREA RATIO=  .942    DELTA=24.9 DEG
GATE OPENING   :   25.00 CM              DUCT VERT. HEIGHT    66.46 CM
GAP            :  105.07 CM              CLEARANCE RATIO      :   .107

                  APERTURE AREA                :   10256.02 CM**2
                  APERT. AREA RATIO (E./A.)    :    1.000 (DEFAULT)
2000 SAMPLES        RADIUS  32.40 CM        ROTOR HEIGHT 62.40 CM

                  UP.HD.      DOWN.HD.     ANG.VEL.      TORQUE
                   (CM)         (CM)        (RPM)        (N.M)
          AVG     97.15        94.88        30.07          .92
          LOW     97.09        94.84        29.68        -3.67
         HIGH     97.19        94.91        30.51         5.80
           AVE. OF ALL LOW  PEAKS    -1.99:    50 VALUES
           AVE. OF ALL HIGH PEAKS     3.73:    52 VALUES

                    HIGH PEAKS                        LOW PEAKS
MAX.                   5.80                             .55
TOP RANGE AVE          4.81:    19 VALUES               .04:     8 VALUES
MID RANGE AVE          3.54:    23 VALUES             -1.79:    26 VALUES
LOW RANGE AVE          2.14:    10 VALUES             -2.94:    18 VALUES
MIN.                   1.18                           -3.67

25% SIG. LEVEL         5.00:    13 VALUES             -3.12:    13 VALUES
SIG. LEVEL RATIO       5.44                           -3.40
NB OF PEAK/REV         5.19                            4.99
PK TO PK (SIG. VALUES) RATIO TO MEAN     8.835
TORQUE STANDARD DEVIATION                2.181

                  EXTRACTION  (RHO*G*Q*H)                .014
                  POWER COEFF. (CK)                     .039
                  POWER COEFF. (CP)                     .124
                  DISCHARGE COEFF. (CD)                1.281
                  TIP SPEED RATIO (RW/VA)              1.215
                  TURBINE POWER                         4.45   WATTS
                  TOTAL POWER DISSIPATED              329.28   WATTS
                  CALCULATED TURBINE EFF.               4.66   %
                  UPSTREAM VELOCITY (V1)               57.38 (CM/S)
                  APERTURE VELOCITY (VA)               83.96 (CM/S)
                  VA/V1                                1.463
                  VOLUME FLOW RATE                     1.533 ((M**3)/S)
                  TOTAL ENERGY HEAD UPSTREAM           98.83 (CM)
                  TOTAL ENERGY HEAD DOWNSTREAM         96.64 (CM)
                  BLADE VELOCITY                      102.03 (CM/S)
                  RELATIVE VELOCITY                   132.13 (CM/S)
                  REYNOLDS NUMBER-BLADE (NOMINAL CHORD)  75042.
                  REYNOLDS NUMBER-SHAFT                  22431.
                  SPECIFIC SPEED (RPM,KW,M)            237.87
                  POWER LOSS IN BEARINGS ONLY     :      1.55   WATTS
                  ARM DRAG FACTOR (ONE ARM)             .136
```

NUMBER OF PEAKS PER REVOLUTION

RATIO : $\frac{\text{HIGH TORQUE (25\%)}}{\text{AVERAGE TORQUE}}$

AVERAGE OF TOP 25% OF HIGH PEAKS / LOW PEAKS

FIGURE 3.5 Typical computer analysis output

FIGURE 3.6

Calibration runs

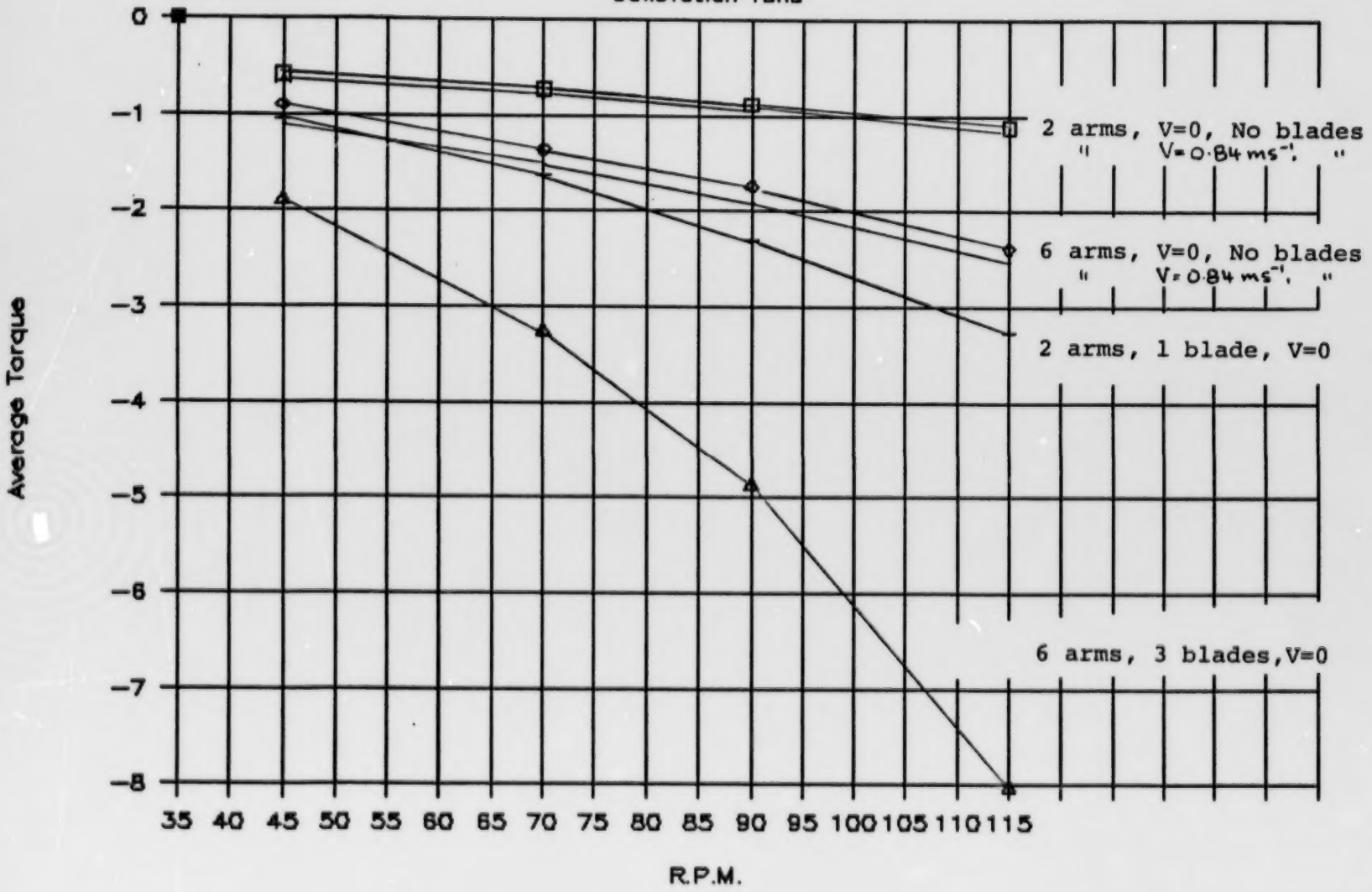


FIGURE 3.7

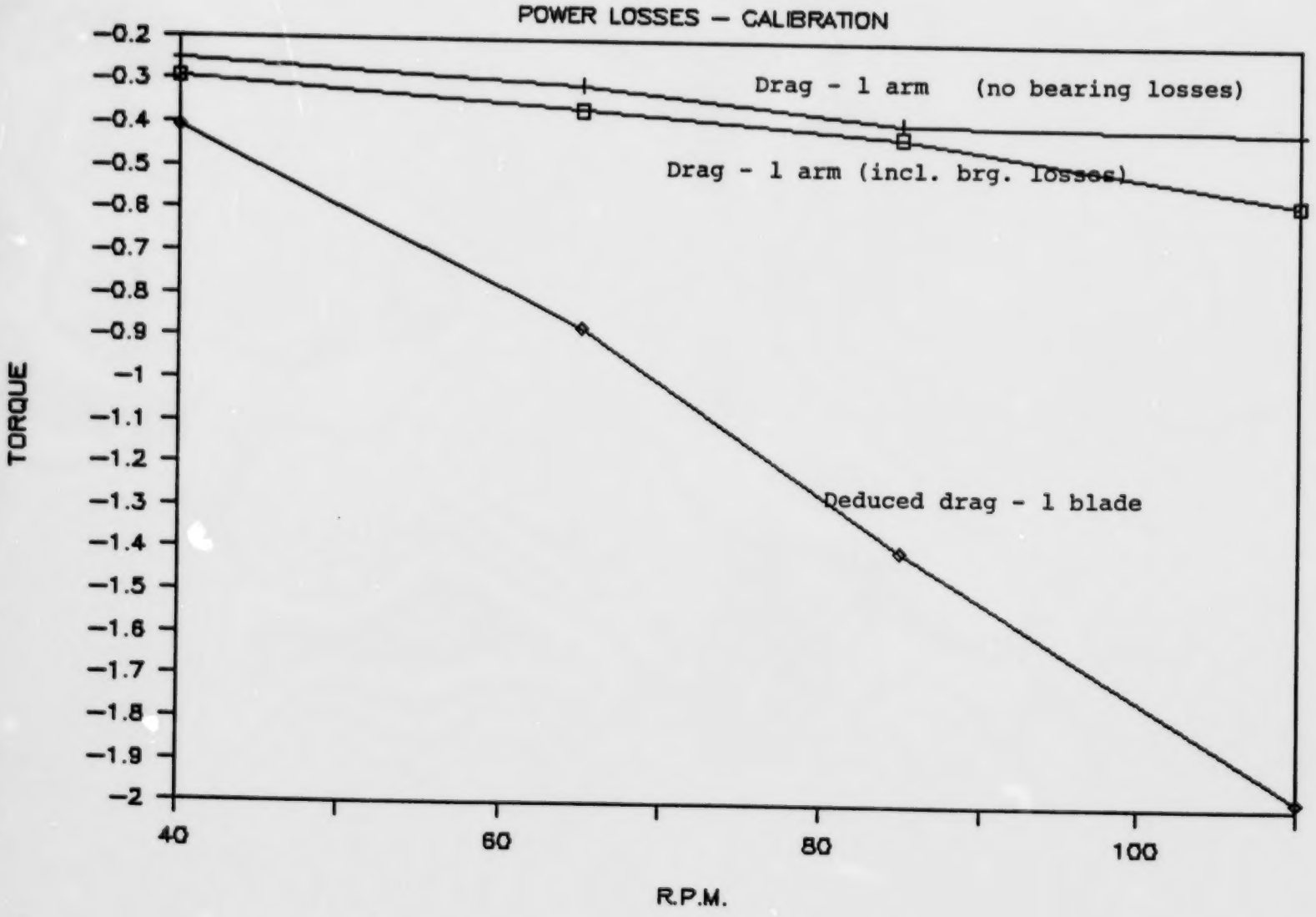

POWER LOSSES – CALIBRATION
Drag - 1 arm (no bearing losses)
Drag - 1 arm (incl. brg. losses)
Deduced drag - 1 blade
TORQUE
R.P.M.
-0.2
-0.3
-0.4
-0.5
-0.6
-0.7
-0.8
-0.9
-1
-1.1
-1.2
-1.3
-1.4
-1.5
-1.6
-1.7
-1.8
-1.9
-2
40
60
80
100

FIGURE 3.8

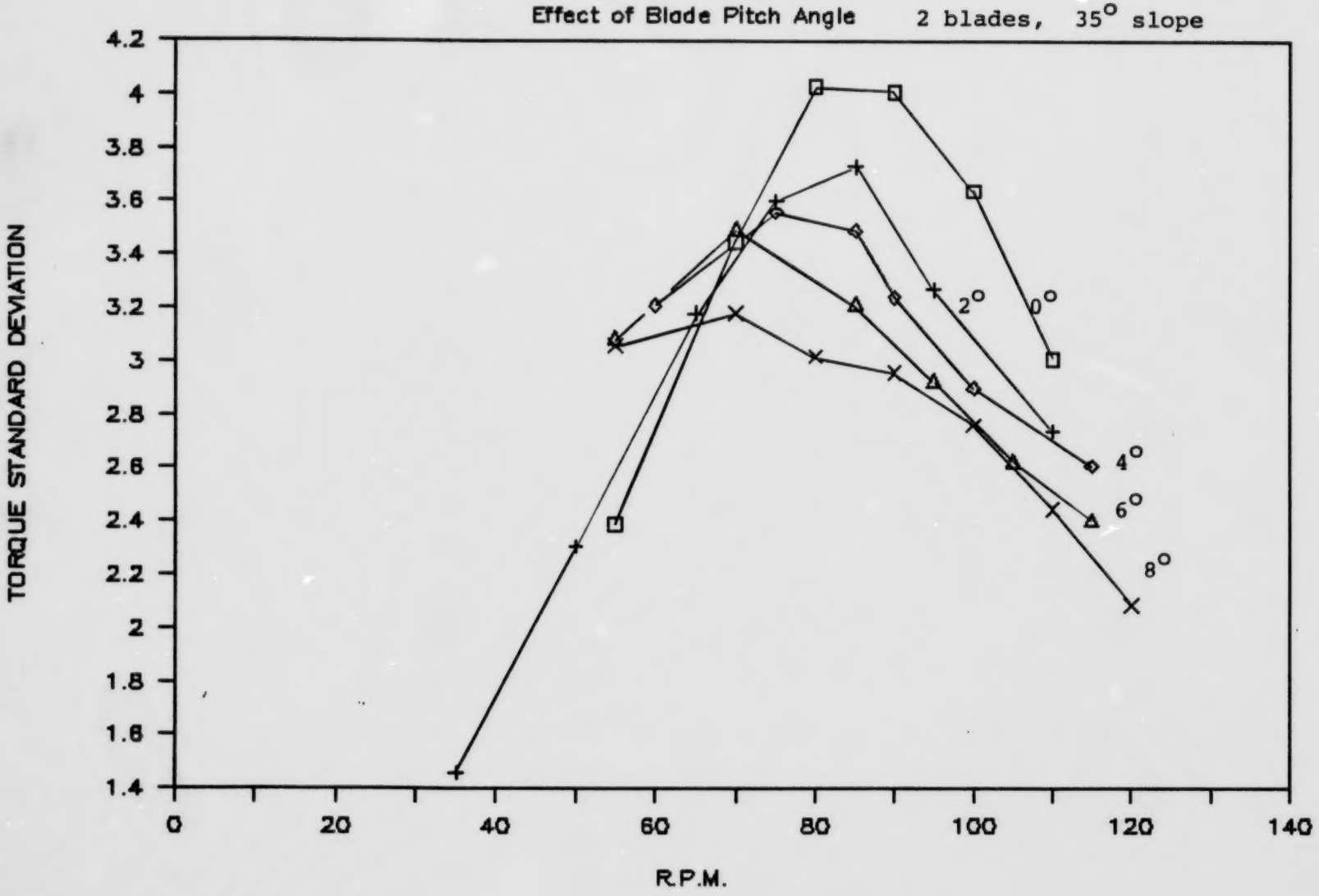

Effect of Blade Pitch Angle
2 blades, 35° slope
TORQUE STANDARD DEVIATION
4.2
4
3.8
3.6
3.4
3.2
3
2.8
2.6
2.4
2.2
2
1.8
1.6
1.4
0
20
40
60
80
100
120
140
R.P.M.
0°
2°
4°
6°
8°

FIGURE 3.9

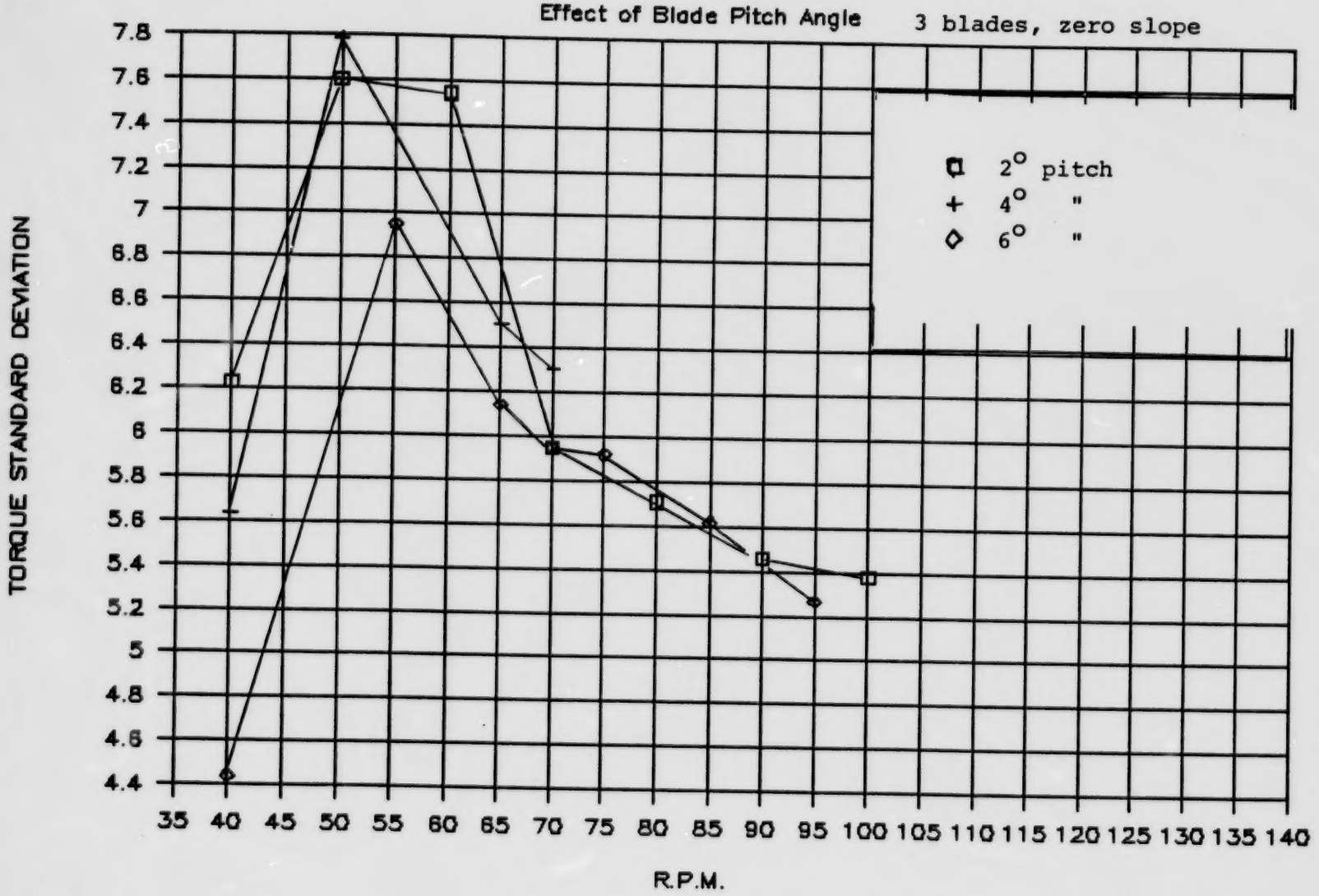

Effect of Blade Pitch Angle
3 blades, zero slope
2° pitch
4° "
6° "
TORQUE STANDARD DEVIATION
7.8
7.6
7.4
7.2
7
6.8
6.6
6.4
6.2
6
5.8
5.6
5.4
5.2
5
4.8
4.6
4.4
35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140
R.P.M.

FIGURE 3.10

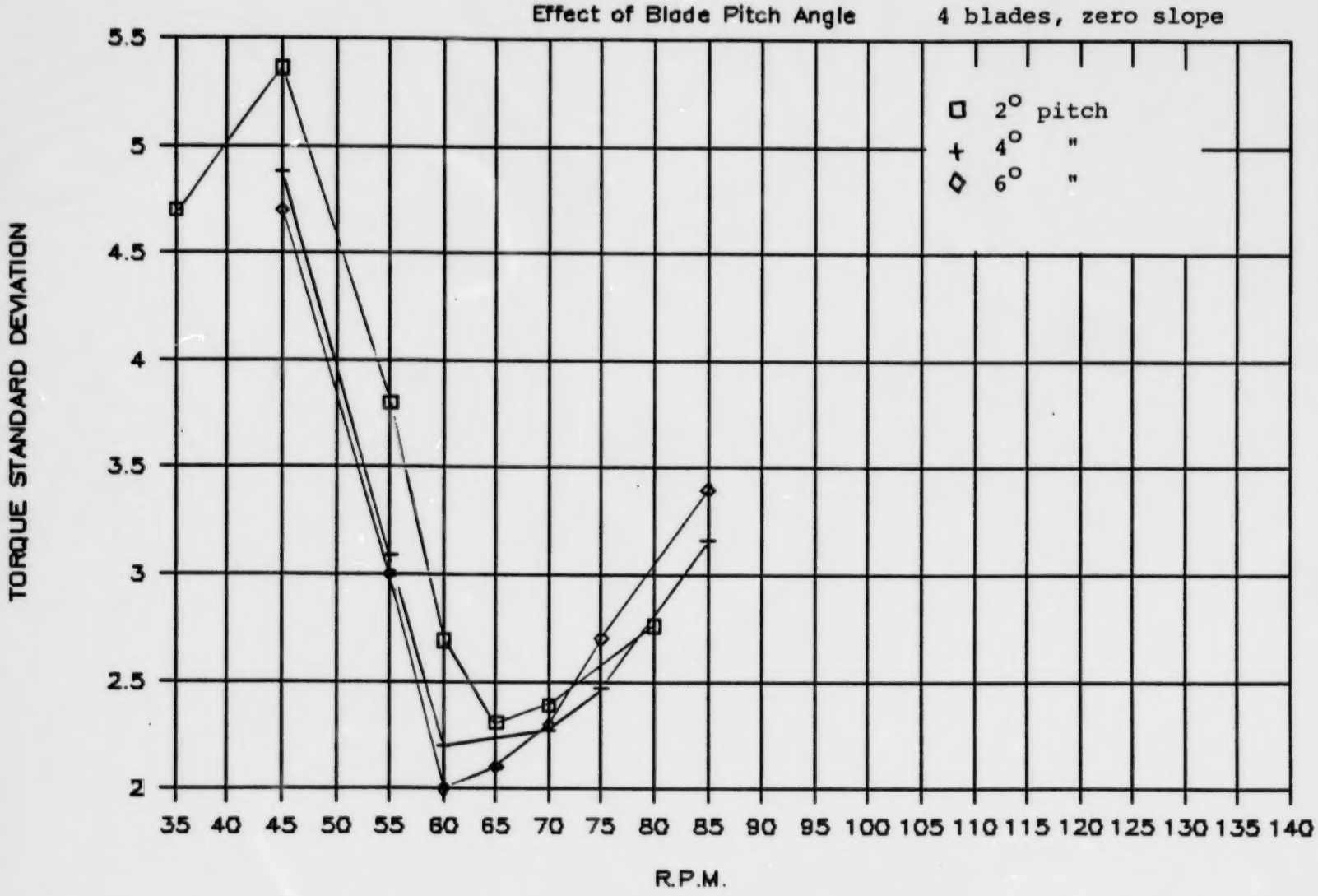

Effect of Blade Pitch Angle
4 blades, zero slope
2° pitch
4° "
6° "
TORQUE STANDARD DEVIATION
5.5
5
4.5
4
3.5
3
2.5
2
35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140
R.P.M.

FIGURE 3.11

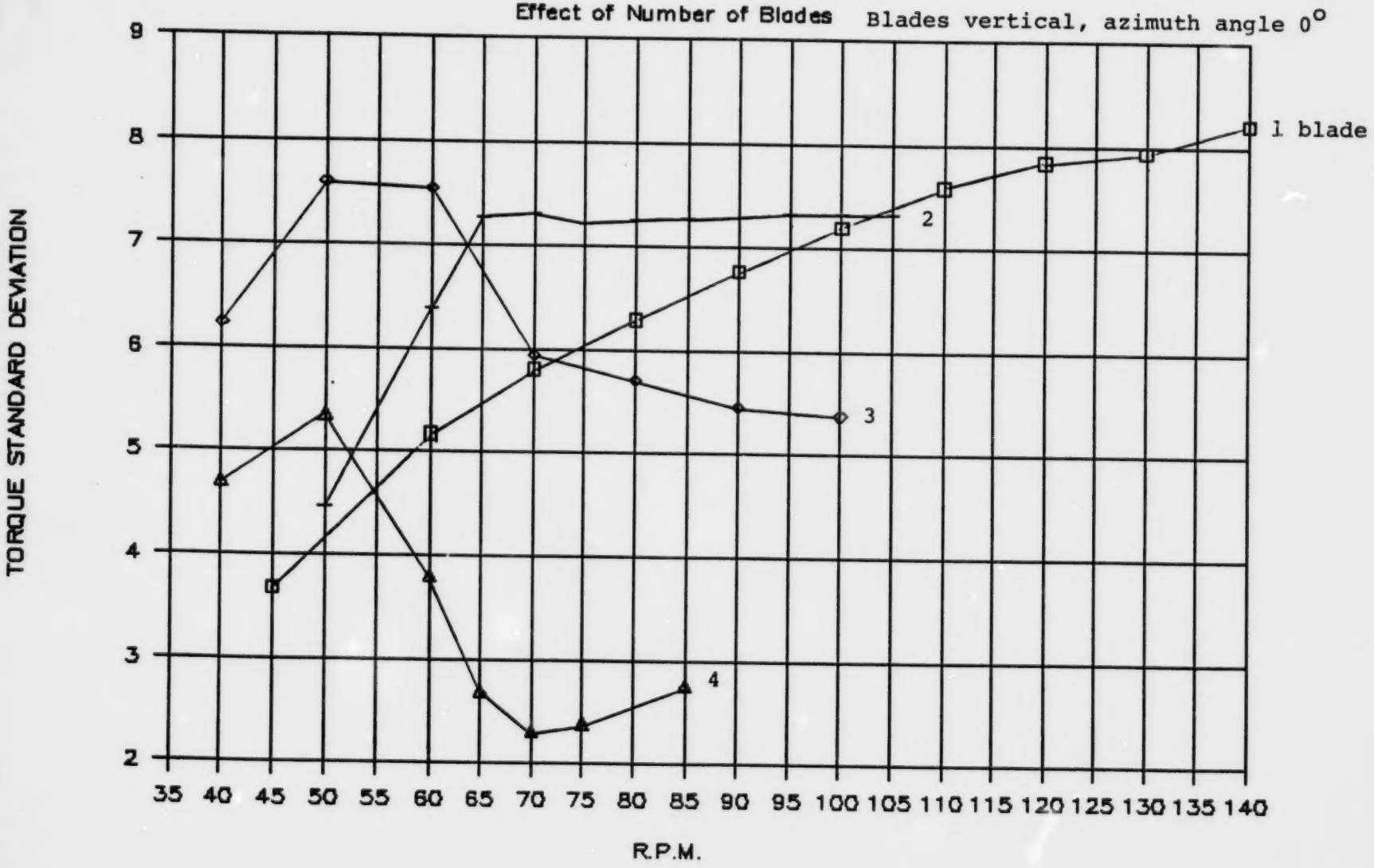

Effect of Number of Blades
Blades vertical, azimuth angle 0°
TORQUE STANDARD DEVIATION
9
8
7
6
5
4
3
2
1 blade
2
3
4
35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140
R.P.M.

FIGURE 3.12

Effect of Number of Blades

Blade slope 9°

Azimuth angle 15°

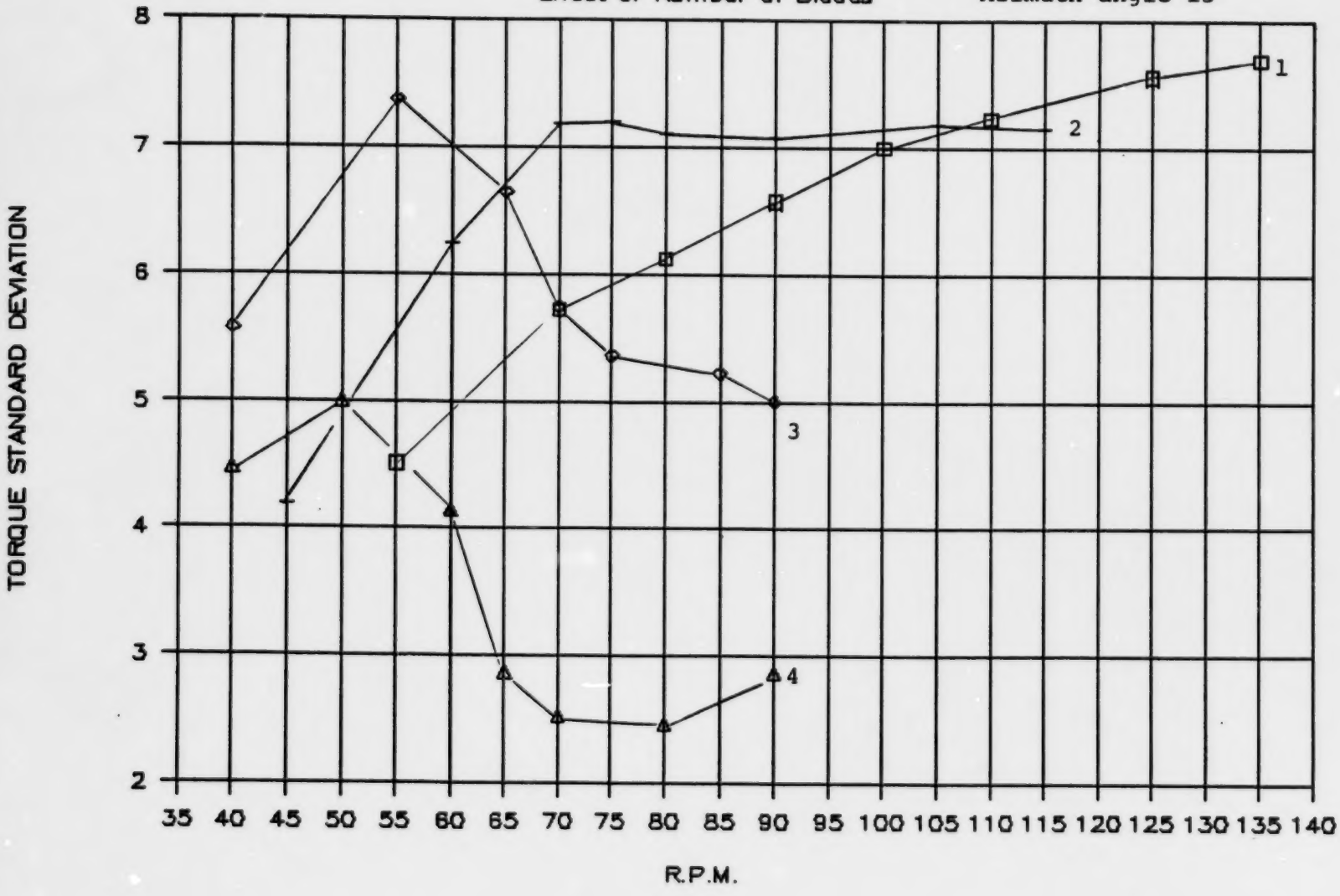


FIGURE 3.13

Effect of Number of Blades

Blade slope 16°
Azimuth angle 30°

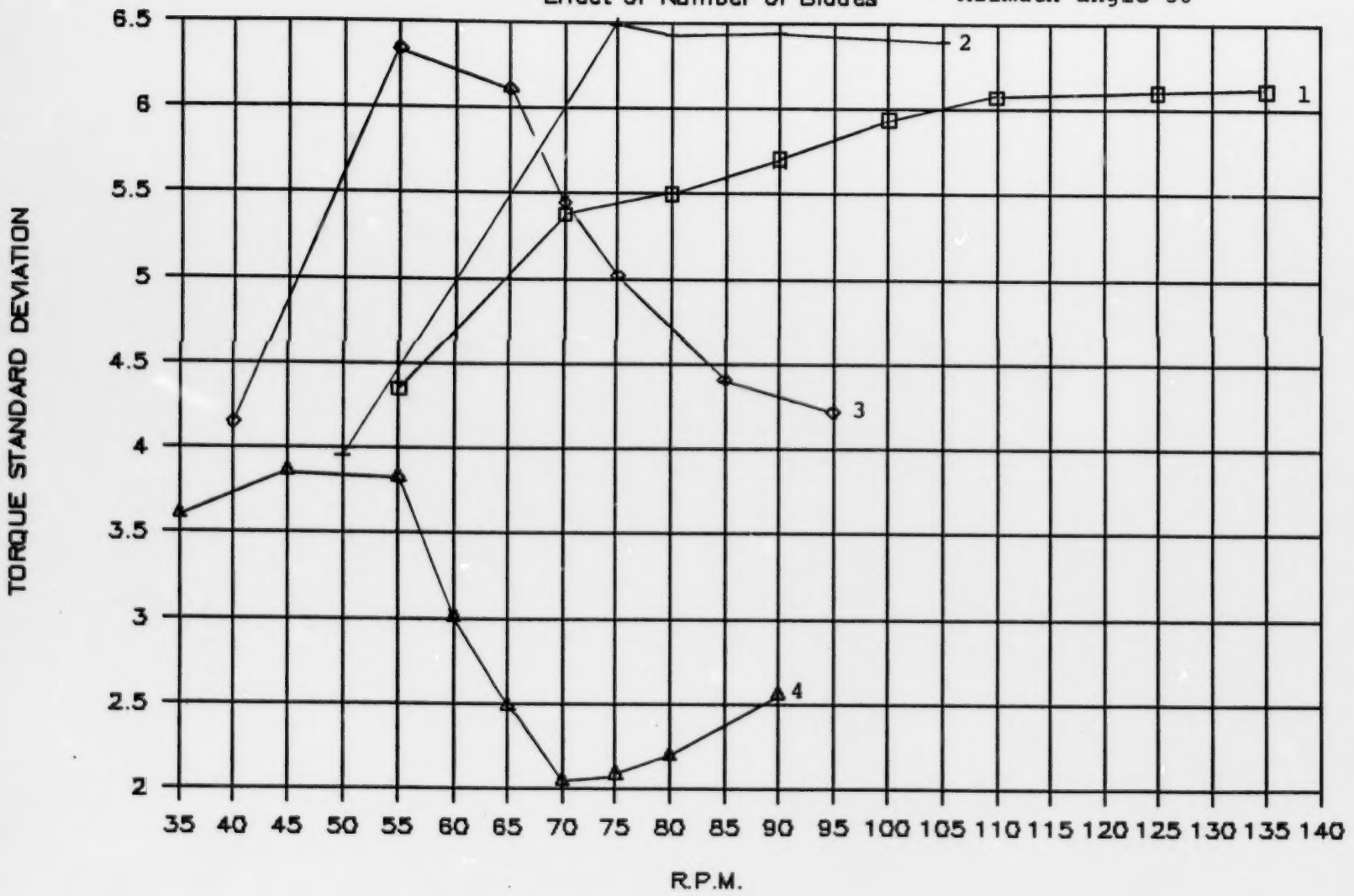


FIGURE 3.14

Effect of Number of Blades

Blade slope 25°
Azimuth angle 50°

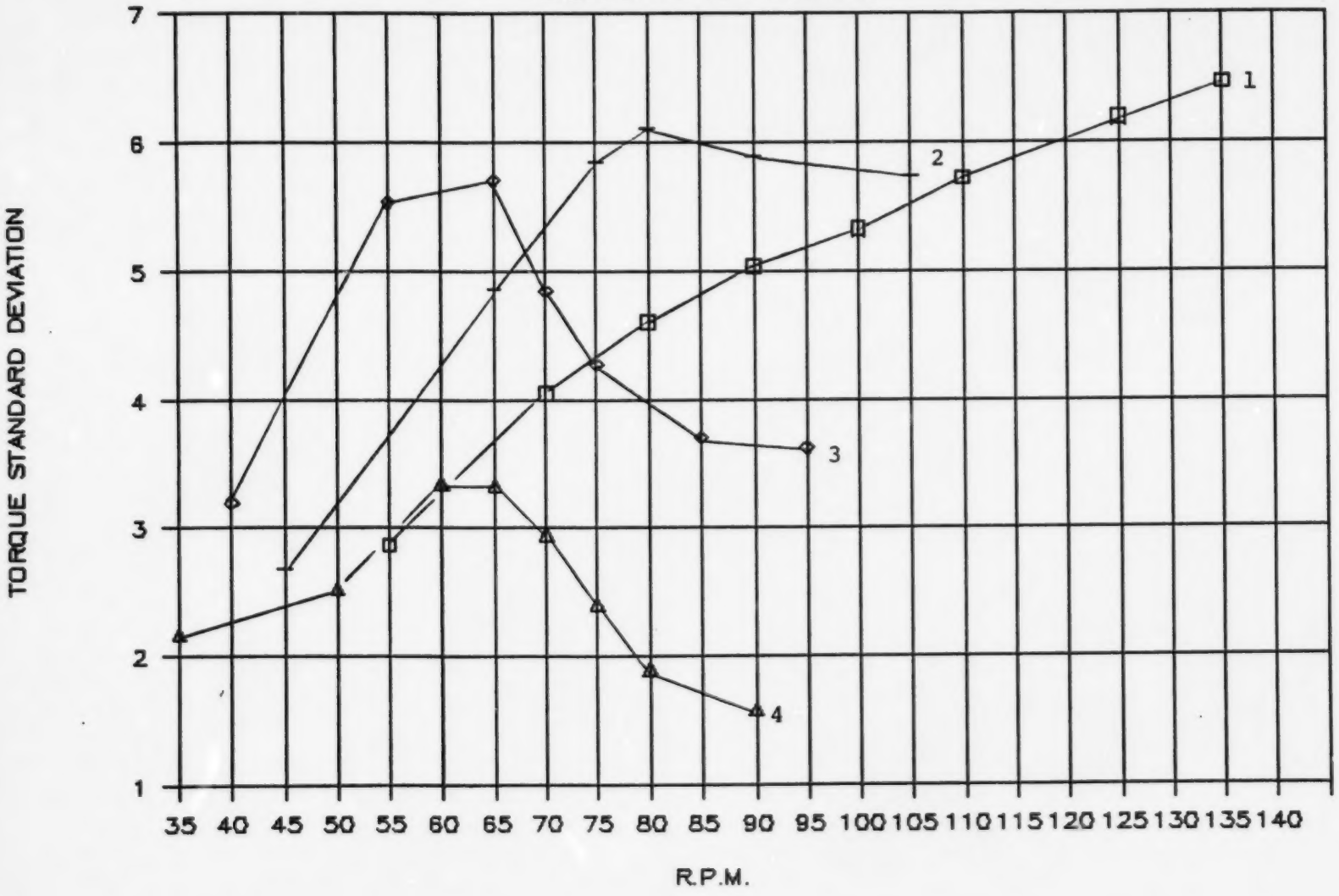


FIGURE 3.15

Effect of Number of Blades

Blade slope 35°

Azimuth angle 80°

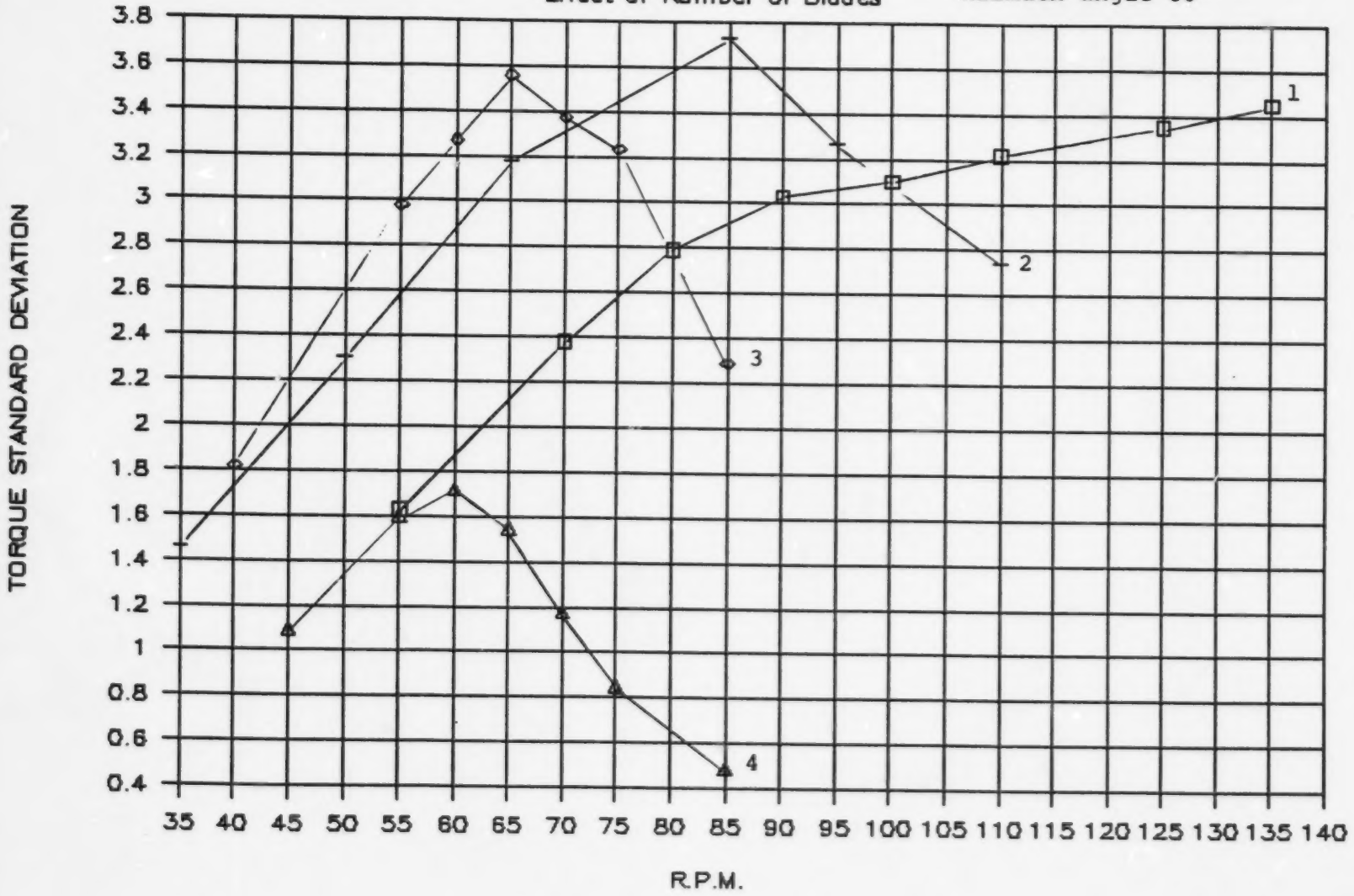


FIGURE 3.16

Effect of Blade Slope

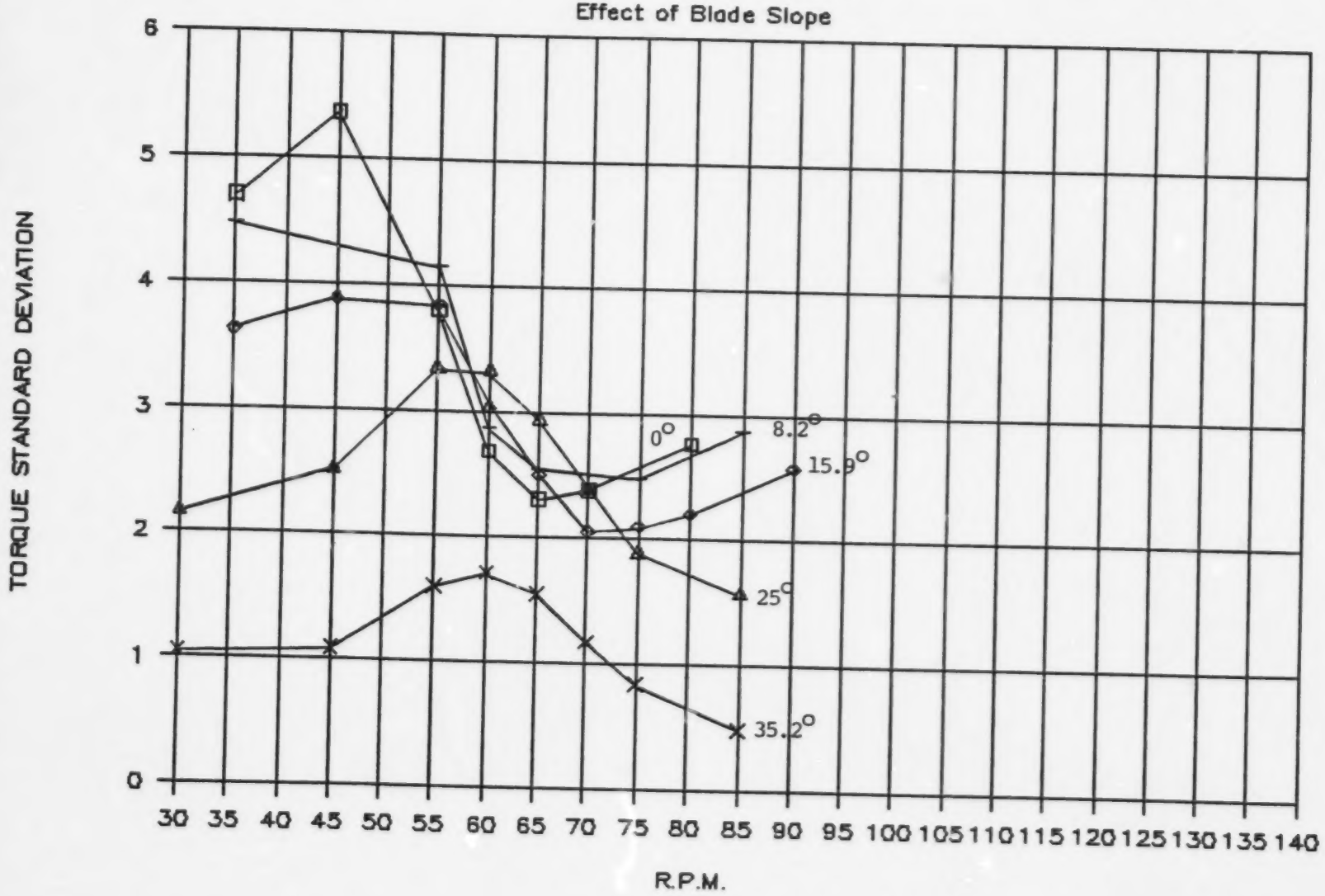


FIGURE 3.17

Effect of Number of Blades

Zero blade slope

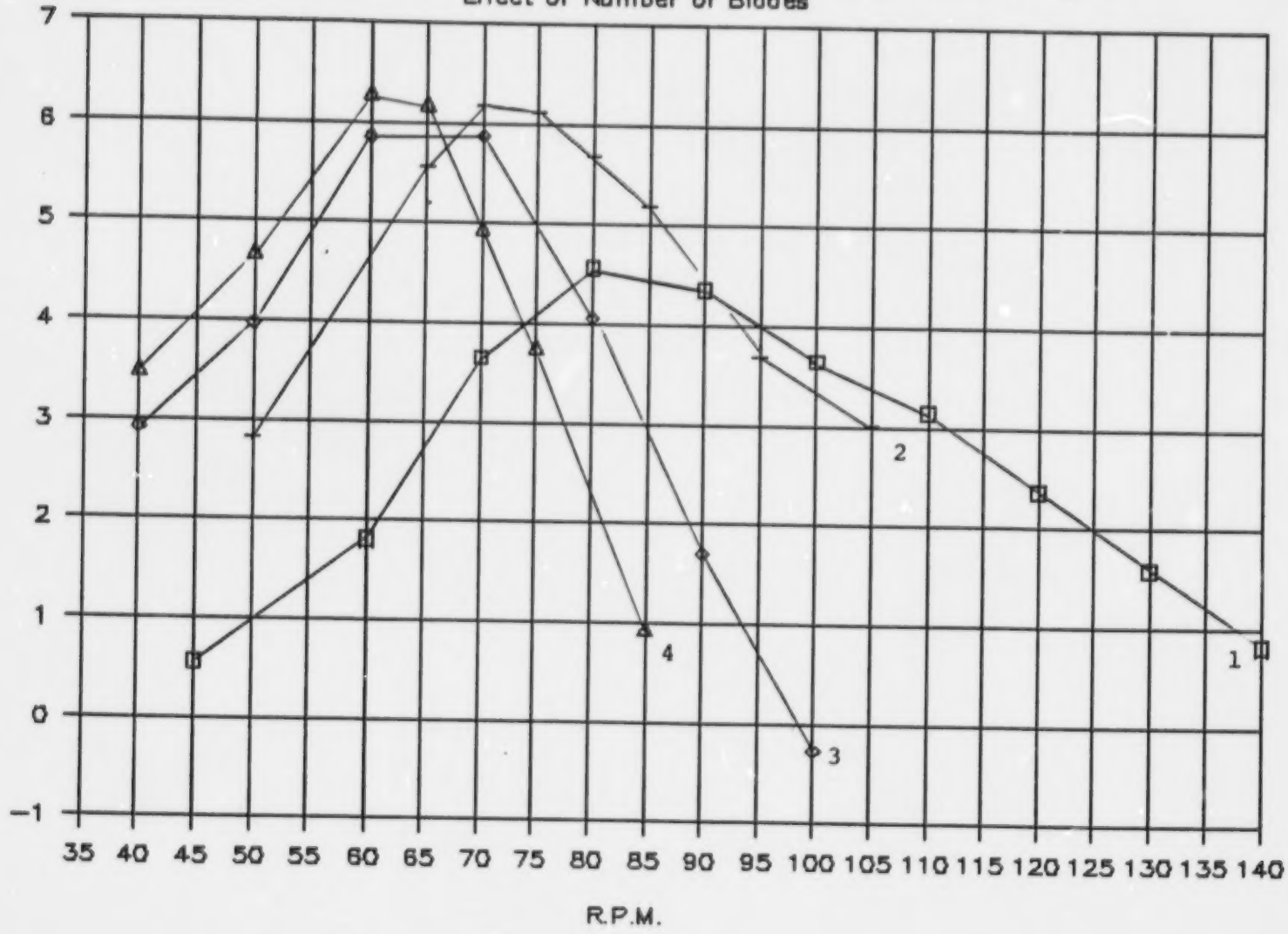


FIGURE 3.18

Effect of Number of Blades Approx. 9° blade slope

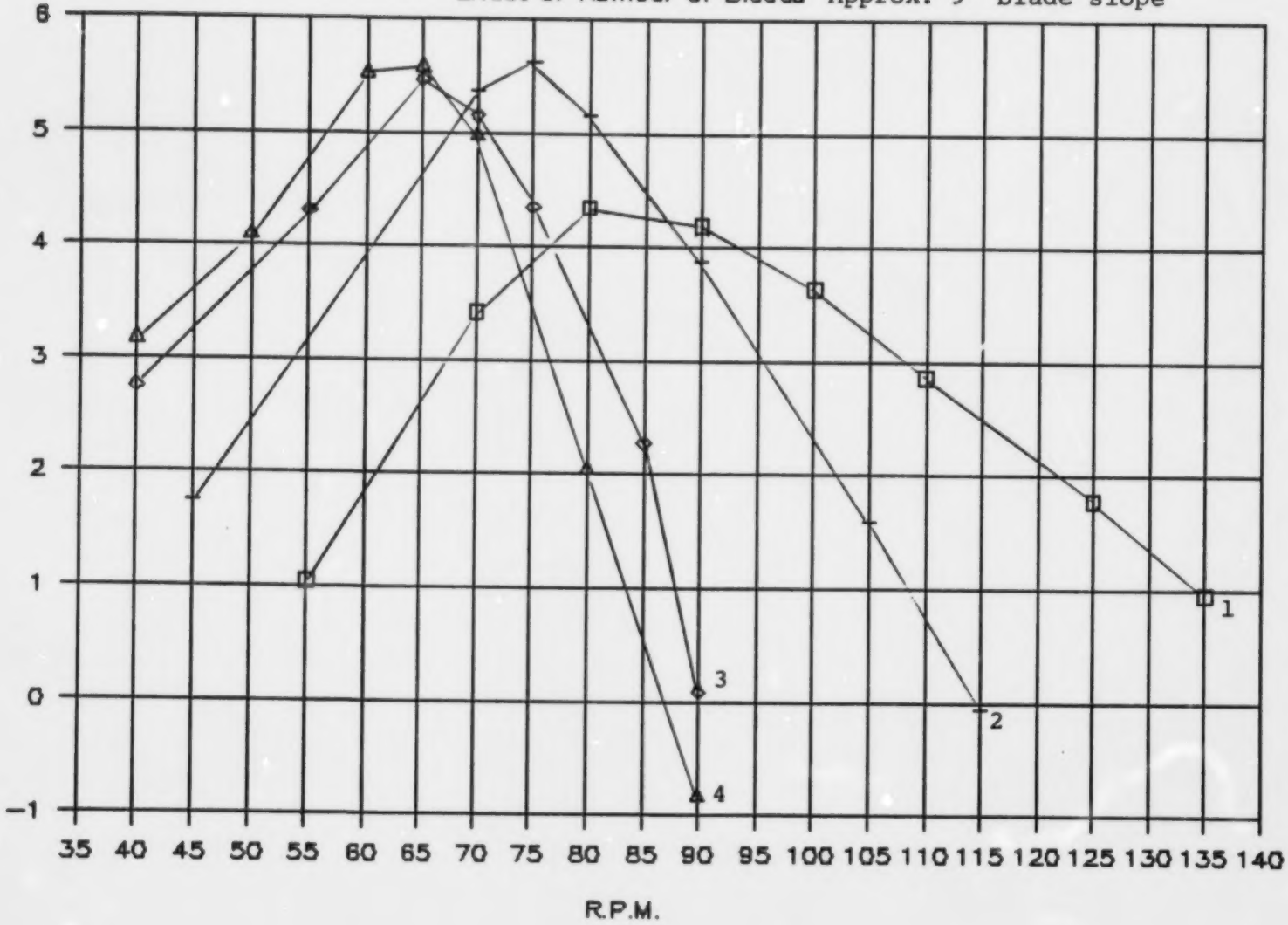


FIGURE 3.19

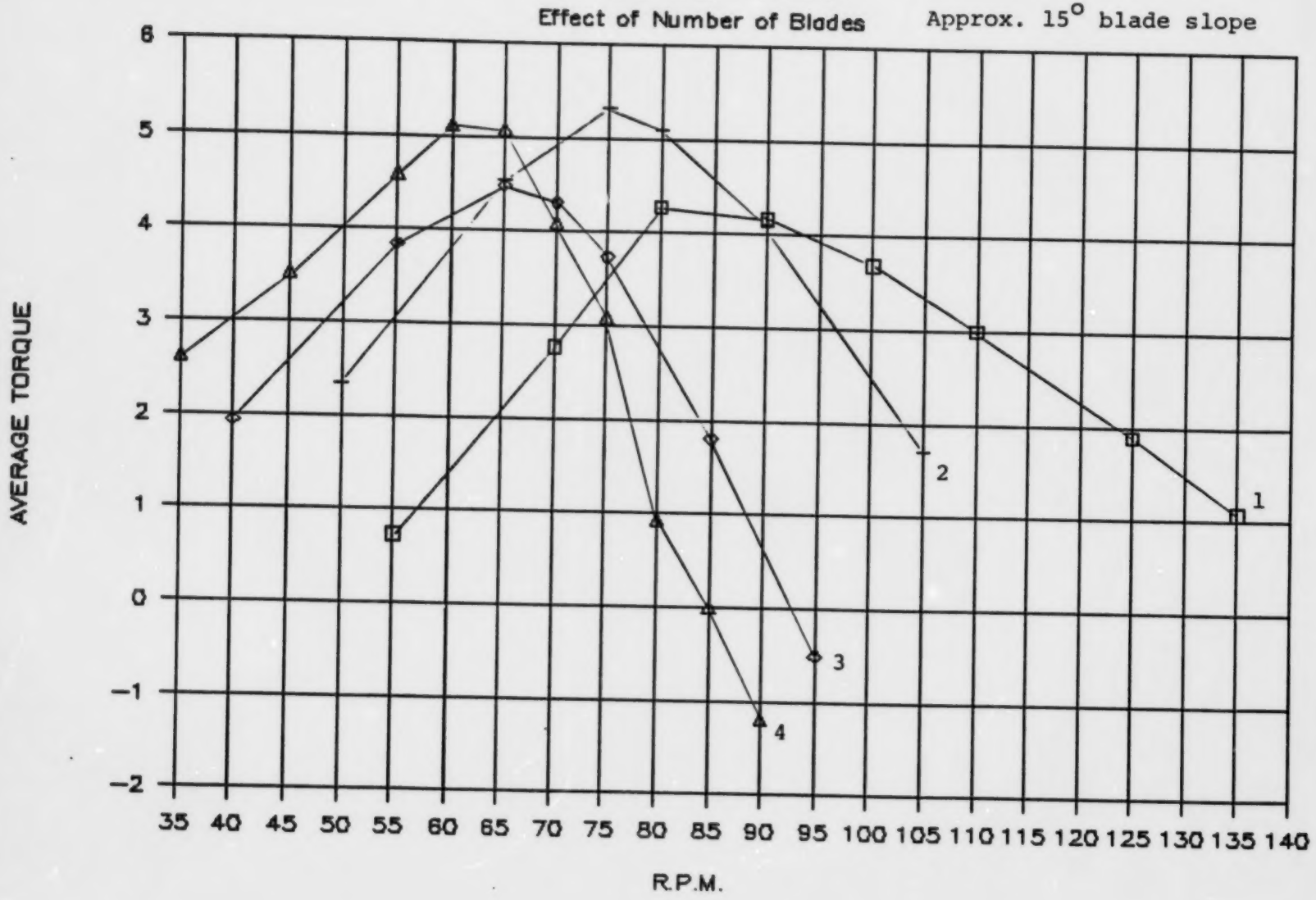

Effect of Number of Blades
Approx. 15° blade slope
AVERAGE TORQUE
6
5
4
3
2
1
0
−1
−2
35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140
R.P.M.
1
2
3
4

FIGURE 3.20

Effect of Number of Blades Approx. 25° blade slope

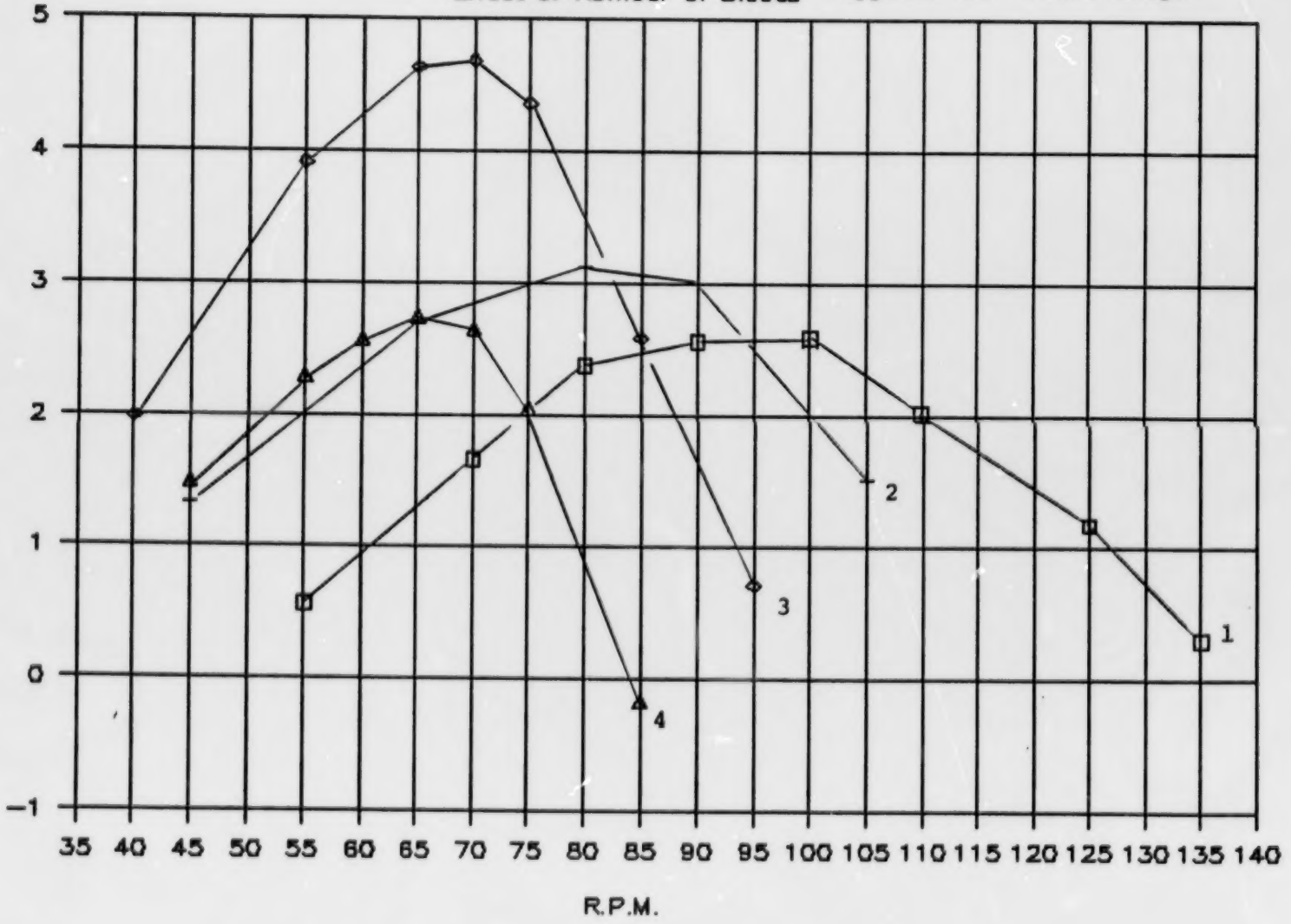


FIGURE 3.21

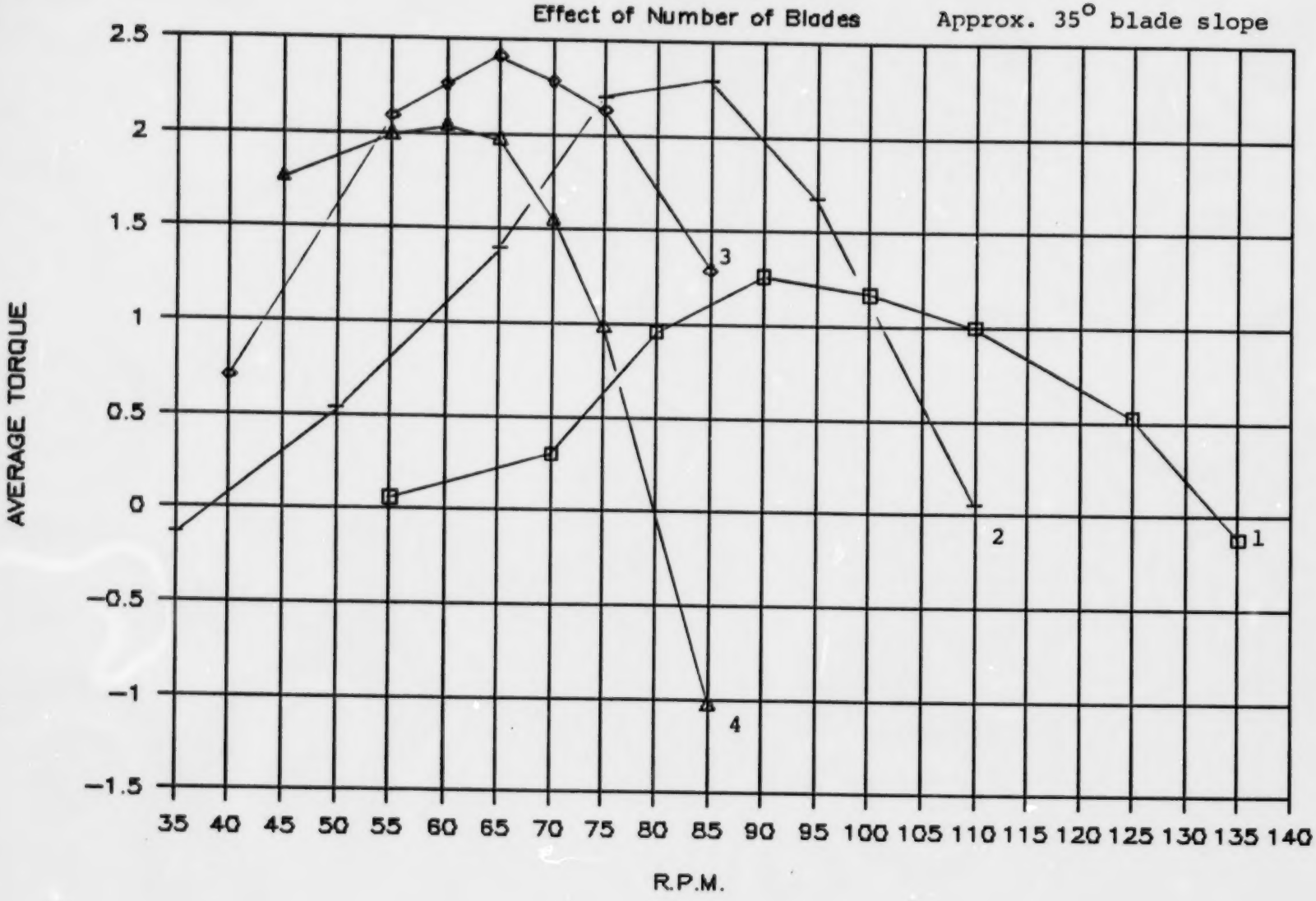

Effect of Number of Blades
Approx. 35° blade slope
AVERAGE TORQUE
2.5
2
1.5
1
0.5
0
−0.5
−1
−1.5
35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140
R.P.M.
1
2
3
4

FIGURE 3.22

Effect of Blade Slope

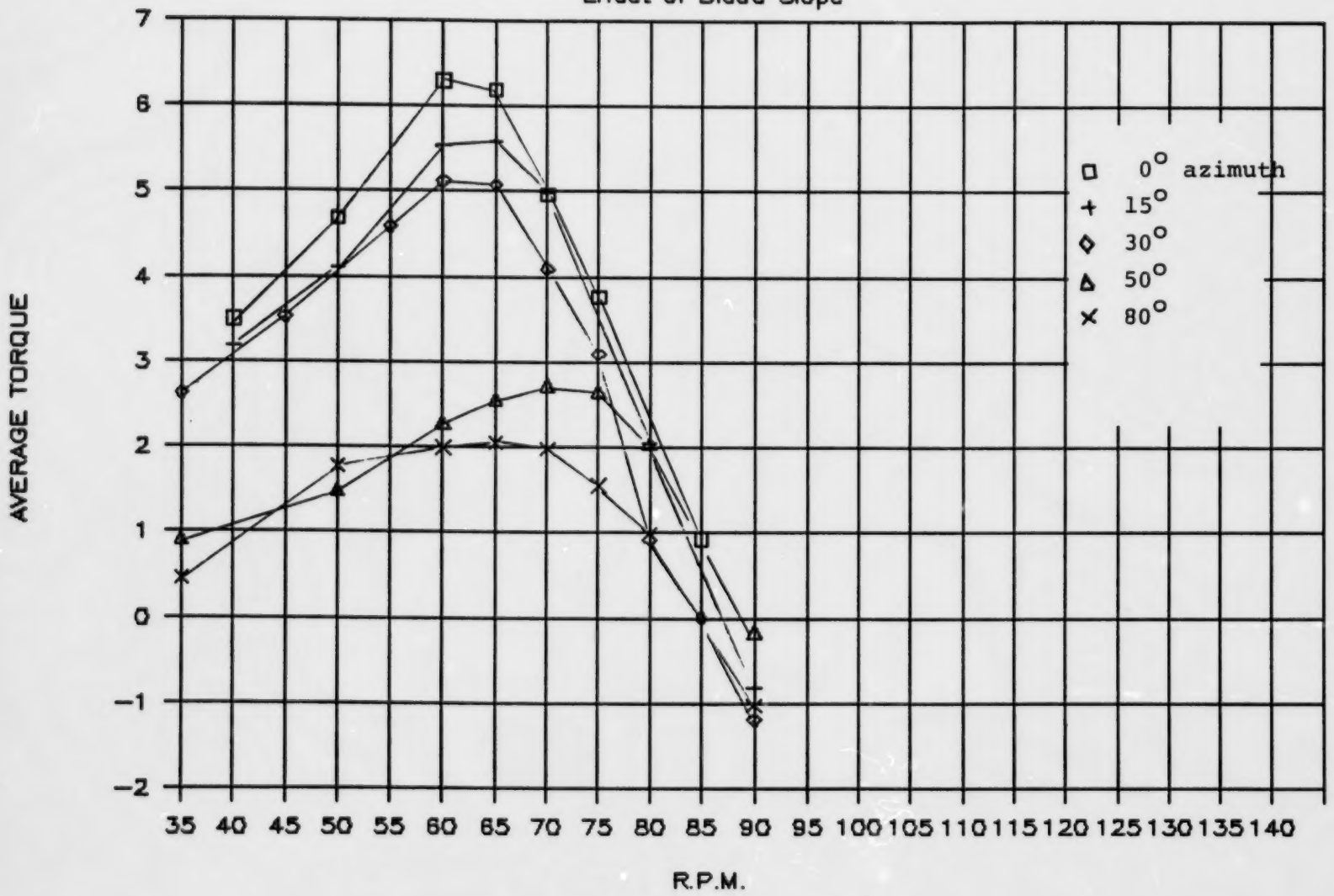
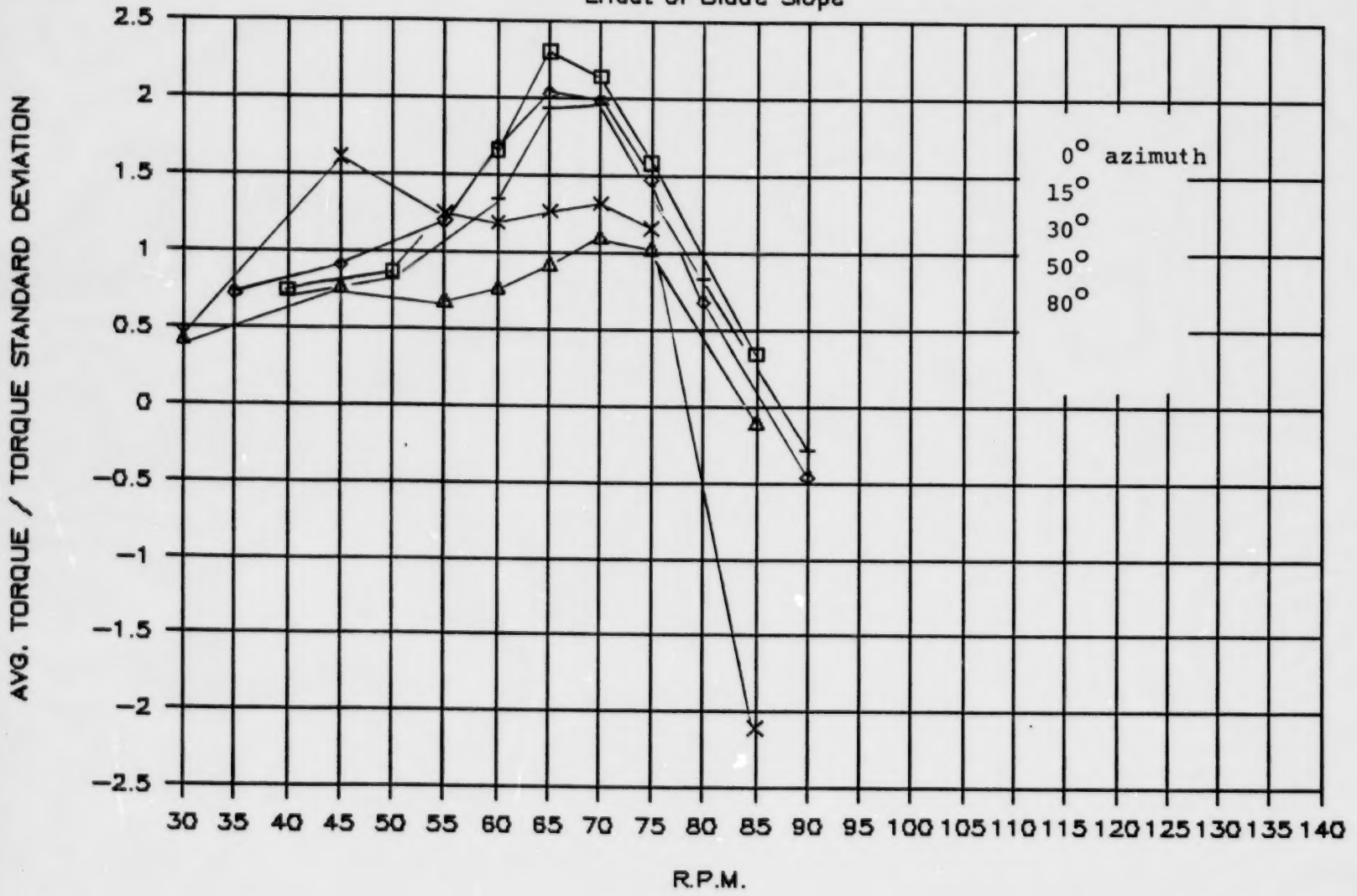


FIGURE 3.23

Effect of Blade Slope



4.0 COMPUTER PREDICTION OF TURBINE PERFORMANCE

4.1 Modelling approach

An analysis of a vertical axis turbine having sloped blades was undertaken to determine the effects of blade slope on the torque output of a turbine. Results were obtained for a typical turbine having 1, 2, 3 and 4 blades and blade slopes of about 0, 8.5, 15, 25 and 35° from the vertical axis. Because a straight sloped blade must also have a twist in it, stress calculations were also carried out to assess the effect of twist on blade stiffness.

The modelling approach made extensive use of the existing computer program developed by GasTOPS Ltd. for vertical axis turbine performance analysis. The program predicts blade loads and torques as well as the overall performance of a crossflow turbine.

Because the GasTOPS program has no facility for modelling sloped blades, Highquest developed a methodology in which the sloped blades were approximated to several discrete segments where each segment was offset from the adjacent segment (Figures 4.1 and 4.2). Blade loads and torques were then calculated for each segment of the blade.

Overall blade loads and torques were obtained by summing the loads and torques on each individual segment. To model a complete blade, two types of segment were used:

end segment	-	includes losses due to tip drag and support arm drag;
mid segment	-	represented a typical section in the middle of a blade (no end losses or support arm).

Validation of this approach is discussed in Section 4.2 below.

4.2 Approximations and sources of error

The GasTOPS program uses a 2 dimensional double multiple streamtube method of analysis, and includes models for blade stall (which apparently contains some errors), blade tip and support arm drag, blade/support arm interference drag and hub wake effects. While giving a good basis of comparison between different turbine geometries, the model makes certain simplifications that do not fully represent a true turbine.

The first of these is the omission of interblade effects, i.e. the effect on downstream blades of upstream blade turbulent wake. Previous validation tests of the program (Ref. 3) show that the simplifications result in differences between actual and theoretically generated results. It is not entirely clear whether these differences are due to the interblade effects or to errors in input data or, indeed, how significant they are.

The second area of likely error which arises particularly when modelling sloped blades is the inability of the model to assess any vertical (or 3rd) dimension of flow. The sloped blades probably impart a vertical component to the streamflow through the turbine which will be negative to the overall performance and which cannot be modelled without a significant redesign of the computer model. This vertical redirection of flow is discussed later in this report.

In addition to these simplifications, NRC report that the blade stall sub-program contains a bug and produces suspect results. Because of this we ran the program with the blade stall option switched off.

We carried out a number of runs to compare the computer generated results with those measured during the course of the experimental work, so doing further validation on the GasTOPS program. These validation runs were compared for a vertical bladed configuration (i.e. with no three dimensional flow effects likely to distort the results).

The purpose of this was to obtain, from the experiments, the actual drag caused by both the blade and support arm sections, as well as the blade tip losses and losses due to the arm/blade interface. It was not possible to run the program to represent all the calibration runs, some of which were carried out with zero water velocity which the program baulked at; it also was unable to accept one of the calibration configurations of support arms and no blades.

However, in the course of these validation runs it became clear that differences in predicted performance due to changes in the inputted drag and loss factors was not representing what was happening in practice (and in fact did not make very significant changes to the results compared with the deviation from practice). The program, however, gave a reasonable representation of the torque pattern though the actual predicted values of torque varied from experiment in some cases by as much as 60%.

We thought that the differences may have been due to the lack of

airfoil data in the GasTOPS program for low Reynold's numbers, and so carried out some comparative runs to compare the torque figures calculated using different airfoil lift and drag data for the three different Reynold's numbers included in the program, but found that this did not change the output significantly.

Despite the lack of confidence we feel in the program's ability to correctly quantify turbine performance, we feel that it is a adequate tool to indicate comparative performance between different turbine geometries.

To verify the general modelling approach used, we compared the results for a "complete" with a "10 segment" straight blade. Figure 4.3 shows that very good agreement was achieved, suggesting that the approach is quite adequate to model sloped blades in this regard.

There was some doubt whether it was valid to extrapolate the results obtained from modelling the very short blade segments, i.e. whether the program could satisfactorily model the 0.0624m segments, so a series of computer runs were undertaken to assess this by modelling blades 10 and 100 times longer and by dividing the results by 10 or 100. Figure 4.3A shows comparisons between segments of 0.0624, 0.624 and 6.24m heights showing that, on torque output, the differences are not significant.

Sloping the blades in effect increases the blade width (chord) and so decreases the apparent blade thickness to chord ratio as seen in the horizontal plane. This effectively changes the blade section, and so has some affect on performance.

In order to assess the approximation of staying with a NACA 0015 75mm blade for our mathematical model when, in practice, the actual blade chord in the plane of rotation was different we ran the program for one set of conditions with:

75mm width	t/c ratio .15	representing 0° slope
76	.148	8
77	.144	16
83	.136	25
92	.123	35

Results are shown in Fig. 4.5.

As the blade slopes the section will no longer be NACA 0015, but will tend towards (but not quite) a NACA 0012 at maximum blade slope, so we also compared calculated performance between turbines with 75mm chord blades of both NACA 0012 and 00125 sections.

In the context of the accuracy of this work we found the differences to be insignificant. We therefore ran the GasTOPS program using a NACA 0015 section throughout, and did not account for the effective change in blade chord caused by the sloped blade.

4.3 Turbine performance prediction runs

In all, nearly 200 runs were computed by the GasTOPS program to model the performance of elements of the test turbine in its different geometrical configurations. A separate program was developed to convert the GasTOPS program data output for the turbine elements to project performance of a "whole" turbine.

Firstly, a series of "calibration" runs was undertaken to assess the effect of changing different parameters on the performance of a full vertical turbine (without sloped blades) of the same aspect ratio as the test turbine used in this work.

As discussed above, we changes values of the drag factors in an attempt to obtain similar theoretical and experimental results, but without significant success. Following this, base case conditions used were:

2 blades	Blade section: NACA 0015		
Pitch angle	2° (toe out)		
50 r.p.m.	Ref. velocity 0.8 m/s		
Blade tip drag:	K1t: -0.03	K2t: 0.01	
Blade/support arm:	Kj: 0.5	Ki: 1	

Parameters varied in turn were:

Number of blades	1 to 5
Pitch angle	-3, -2, -1, 0 1, 2, 4, 5
r.p.m.	20, 40, 60, 80, 100
Ref. velocity	.4 to 1.2 m/s
blade chord	.025 to .2 m.

These calibration runs indicated that a pitch (toe-out) angle of about 4.5° would give optimum performance, and was confirmed later by experimental results.

It was not the purpose of this work to rigorously model the effect of changes in the other variables by a set matrix of modelling runs, and indeed this has been undertaken by others. However, it is clear that some configurations and conditions give much better performance than others. For a turbine with a NACA 0015 blade of chord 75mm (as test turbine), a four bladed machine operating in a 0.84 m/s stream flow appeared to be most

effective at about 75 r.p.m. (tip speed ratio of 2.9).

The main modelling effort was concentrated on modelling the test turbine as actually configured. Results from these computer runs are discussed in Section 4.5 following.

4.4 Stress analysis

4.4.1 General

Blade stresses will be different between sloped and vertically bladed turbines due to the longer overall blade length of a sloped blade turbine and it's less uniform loading. It was thought, however, that the twist in a sloped blade would result in a stiffer blade and as a consequence the stresses would be less despite its increased length. To investigate this, a stress analysis was done to determine the effects of blade slope on the stresses in the blade.

4.4.2 Methodology

The blade stress analysis was done using a standard moment-area technique for determining stresses in a beam. It was assumed the blades are rigidly fixed at each end. To simplify the determination of the blade section moments of inertia, an elliptical cross section was assumed for the blade.

To perform the stress analysis, computer programs were written to read the blade force results generated by the GasTOPS program and to integrate the forces numerically over the length of the blade to obtain the blade deflection curve. Allowance was made in the calculations for the change in the moments of inertia along the blade due to the blade twist. Actual blade stresses were not calculated as deflections were felt to be a more appropriate point of comparison between blades of different slopes.

4.4.3 Results

Blade deflections were calculated for a turbine with the basic geometry of Nova Energy's B1 turbine (Ref. 4) as well as for the actual test turbine used in this project. Blade deflections were calculated for blade slopes of 0 and 45° for the B1 turbine, with results shown in Figures 4.6 and 4.7. These show that the sloped blade has a maximum deflection of approximately one half that of the vertical blade, despite being approximately 25 percent longer. Thus sloping the blade results in an effectively much stiffer blade. Analysis of the actual test turbine support these results. Results of the test turbine stress analysis are

not presented here however, as the predicted deflections were so small (on the order of 0.01 micron) that they were considered insignificant.

4.5 Discussion of results

In the early stages of this project we applied our sloped bladed modelling approach to a turbine representative of Nova Energy's B1 turbine (Refs. 3 & 4) which was tested in the Ottawa River at Cornwall under an NRC sponsored contract. Computer runs were done for a turbine having blade slopes of 0, 15, 30 and 45° and 1, 2, 3 and 4 blades. Net torque curves are given in Figs. 4.8 to 4.15, Figs. 4.12 to 4.15 comparing the torque as a function of the number of blades for a given blade slope.

These results suggested that sloping the blades has some tendency to smooth the torque curve of the turbine but at the expense of reduced average output torque. They also indicated that increasing the number of blades to 4 clearly reduces the magnitude of the torque pulse over other numbers of blades. Significantly the three bladed turbine had particularly large torque pulses.

Subsequent computer work modelled Highquest's test turbine as configured during the experimental runs in NRC's flume tank. Theoretical output could then be compared directly with the experimental results to assess the validity of the modelling approach. A selection of results are shown in Figs. 4.16 to 4.20.

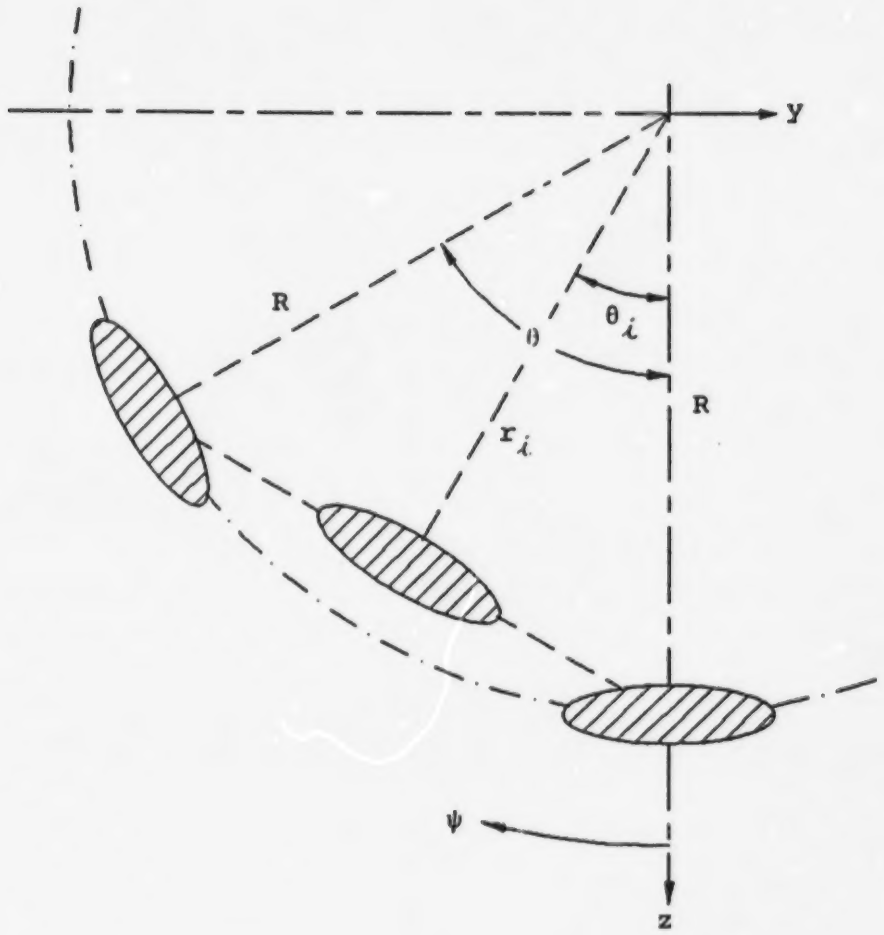


FIGURE 4.1 Cross sectional geometry of sloped blade. Cross sections are taken at the two end points and the mid point of the blade. Y is in the downstream direction.

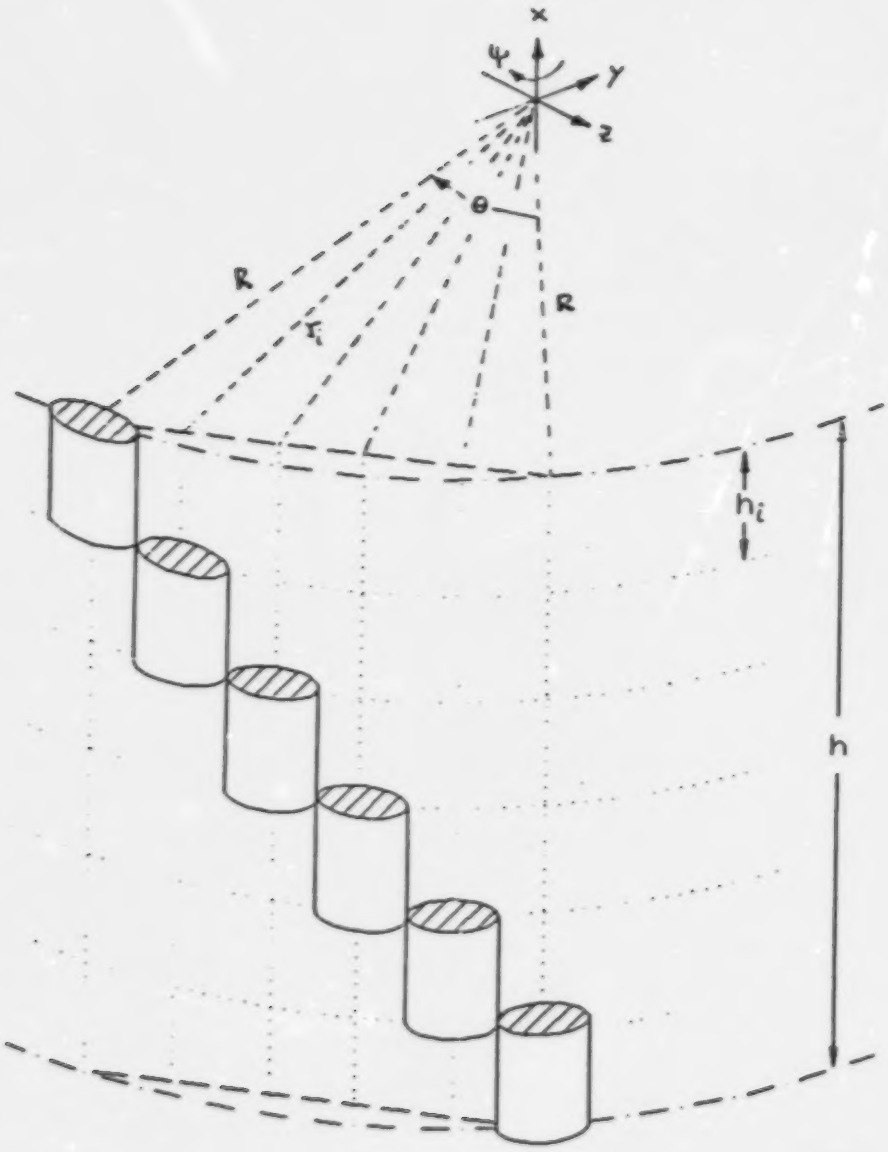


FIGURE 4.2 Typical geometry of segmented model for sloped blade.

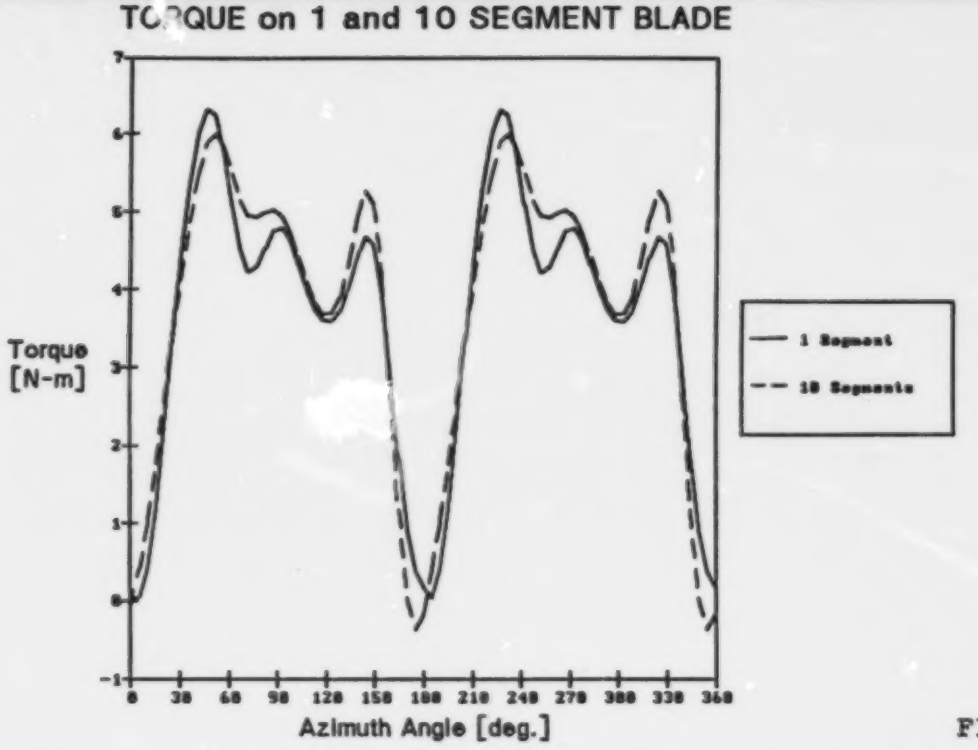
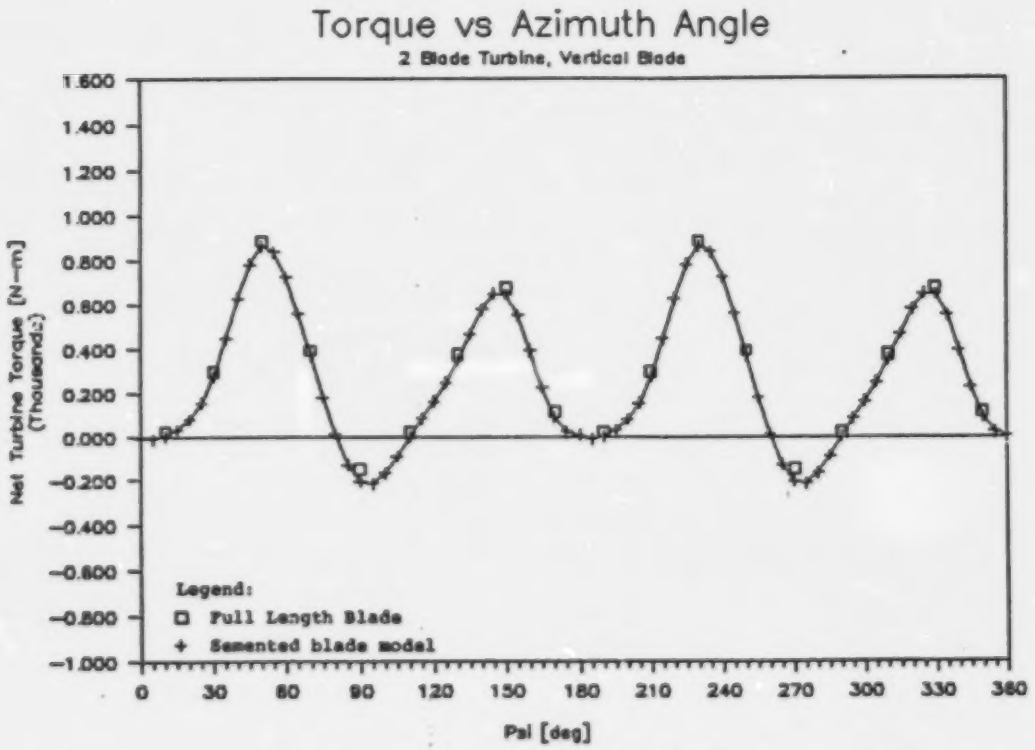


FIGURE 4.3



Comparison of torque curves calculated for 2 blade turbine using full length blade and segmented blade models.

FIGURE 4.3A

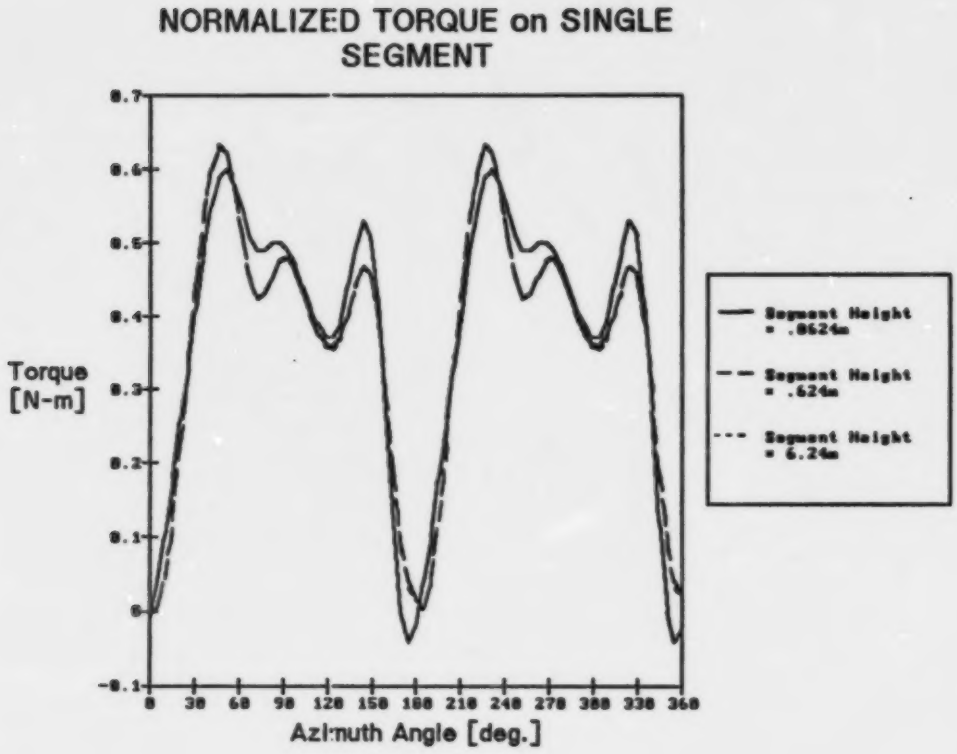
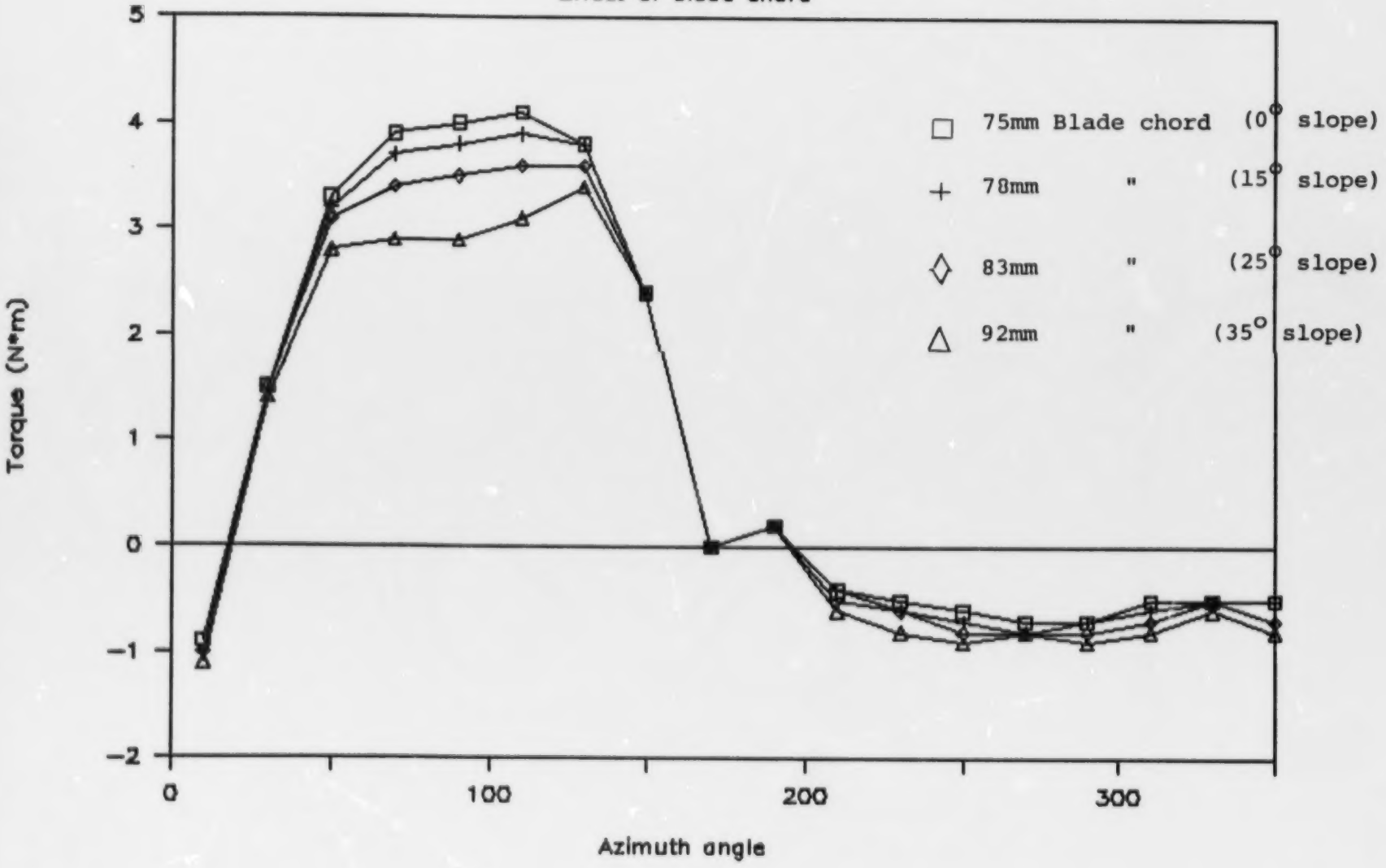

NORMALIZED TORQUE on SINGLE SEGMENT
Torque [N-m]
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
-0.1
0
30
60
90
120
150
180
210
240
270
300
330
360
Azimuth Angle [deg.]
Segment Height = .0624m
Segment Height = .624m
Segment Height = 6.24m

FIGURE 4.4

FIGURE 4.5

Effect of blade chord



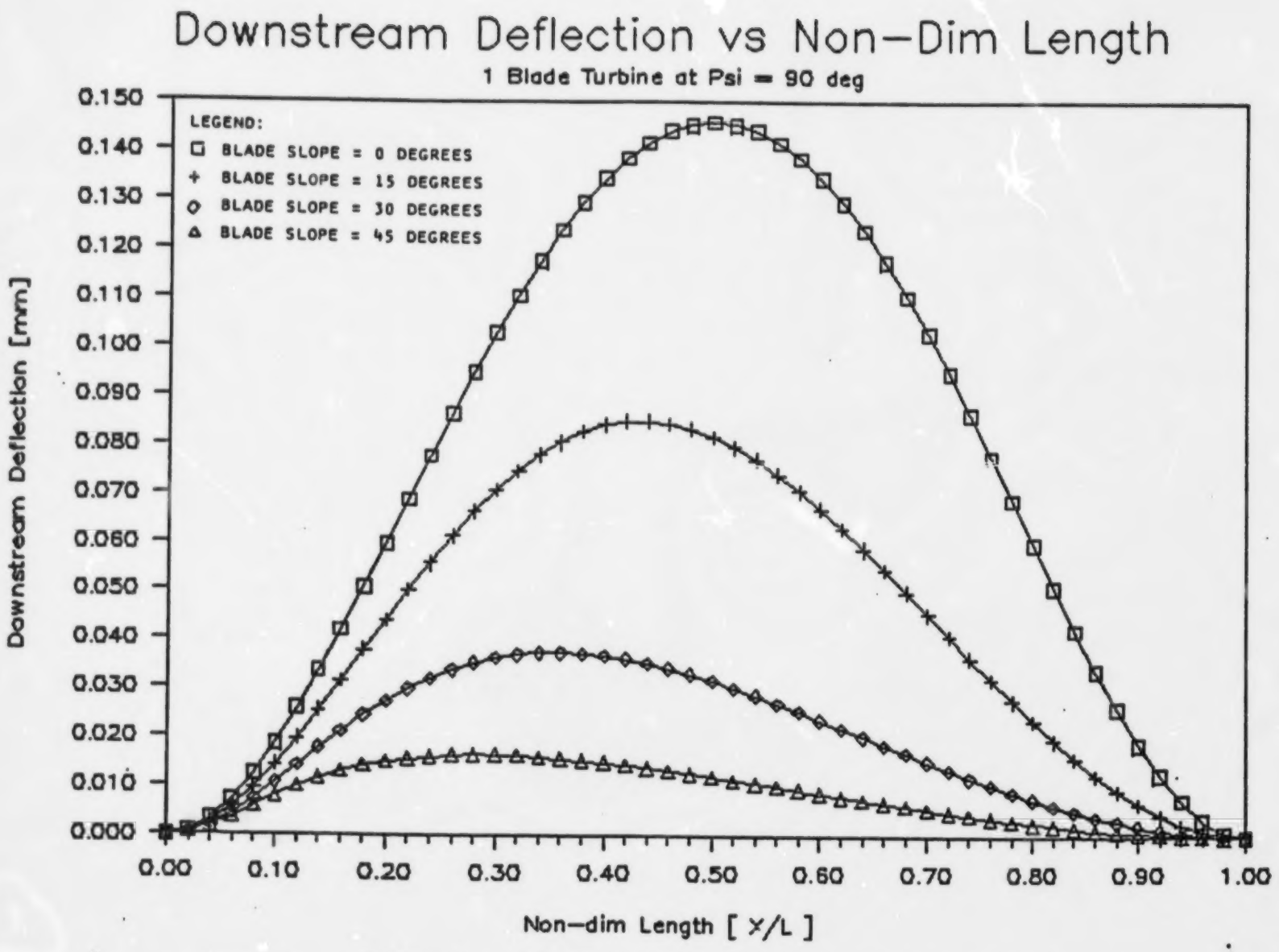


FIGURE 4.6 Deflection of blade having various amounts of blade slope. Deflections are calculated at turbine azimuth angle of 90 degrees.

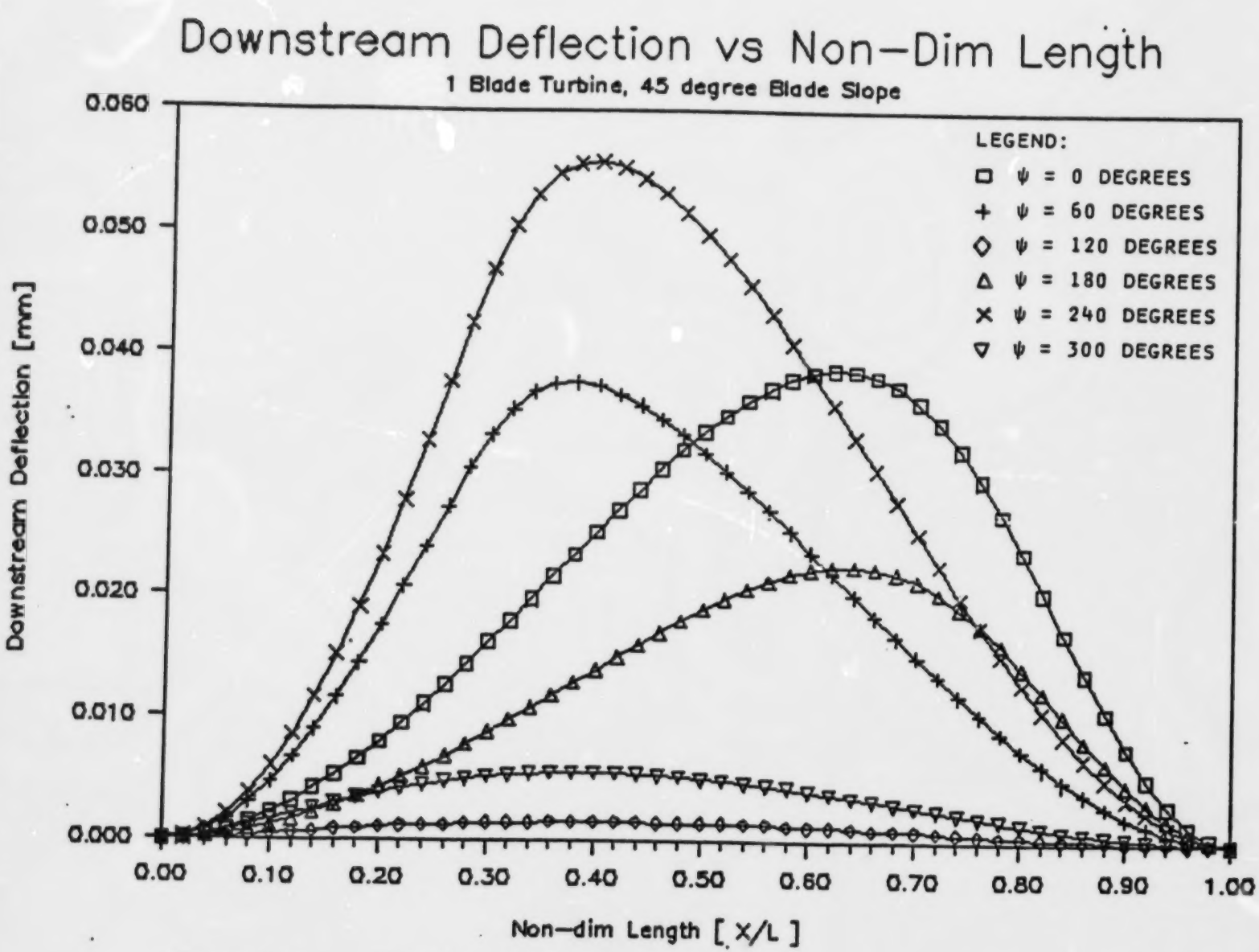


FIGURE 4.7 Deflection of turbine blade having 45 degree slope at various azimuth angles.

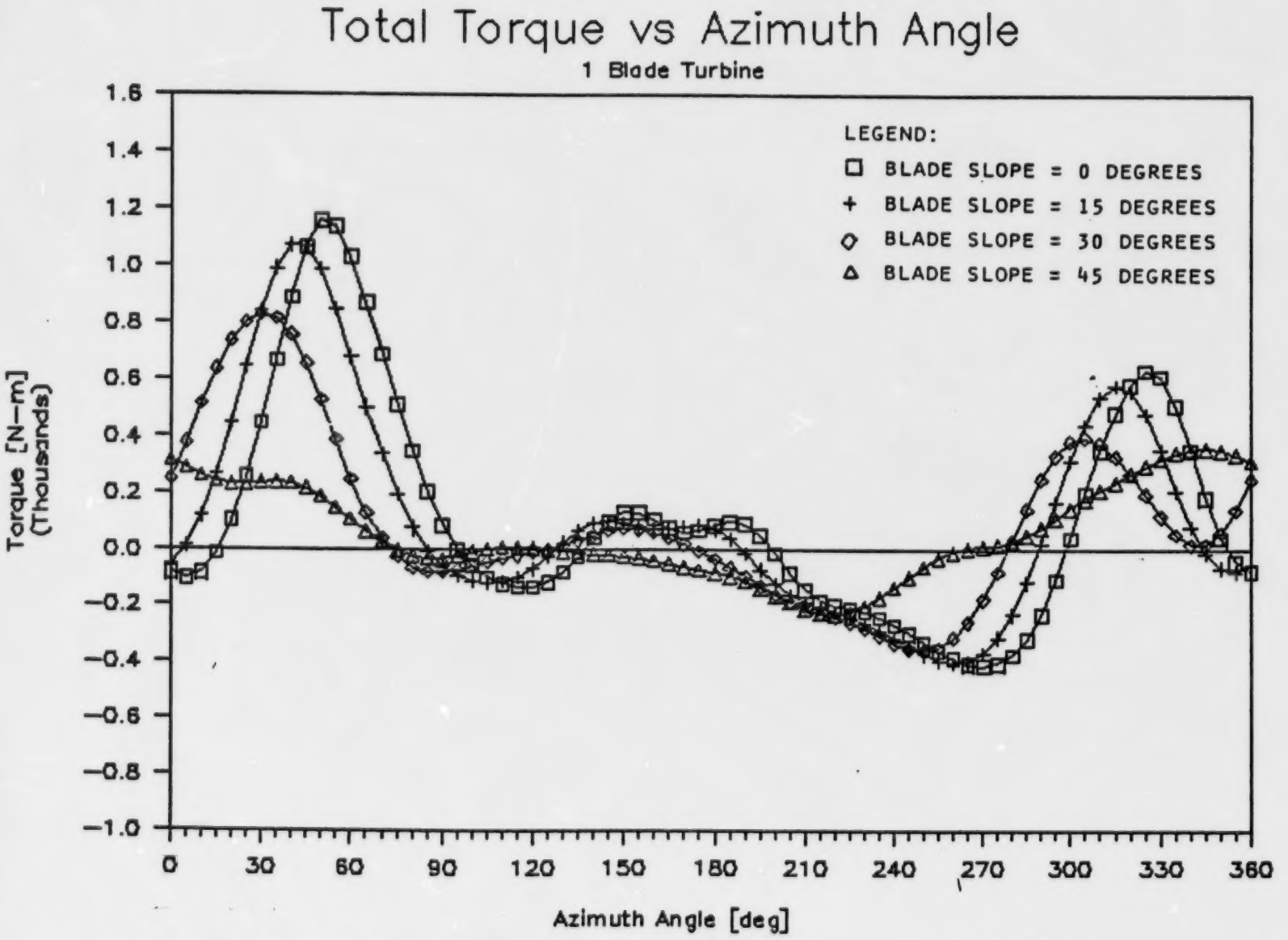


FIGURE 4.8 Torque curves for 1 blade turbine and various blade slopes

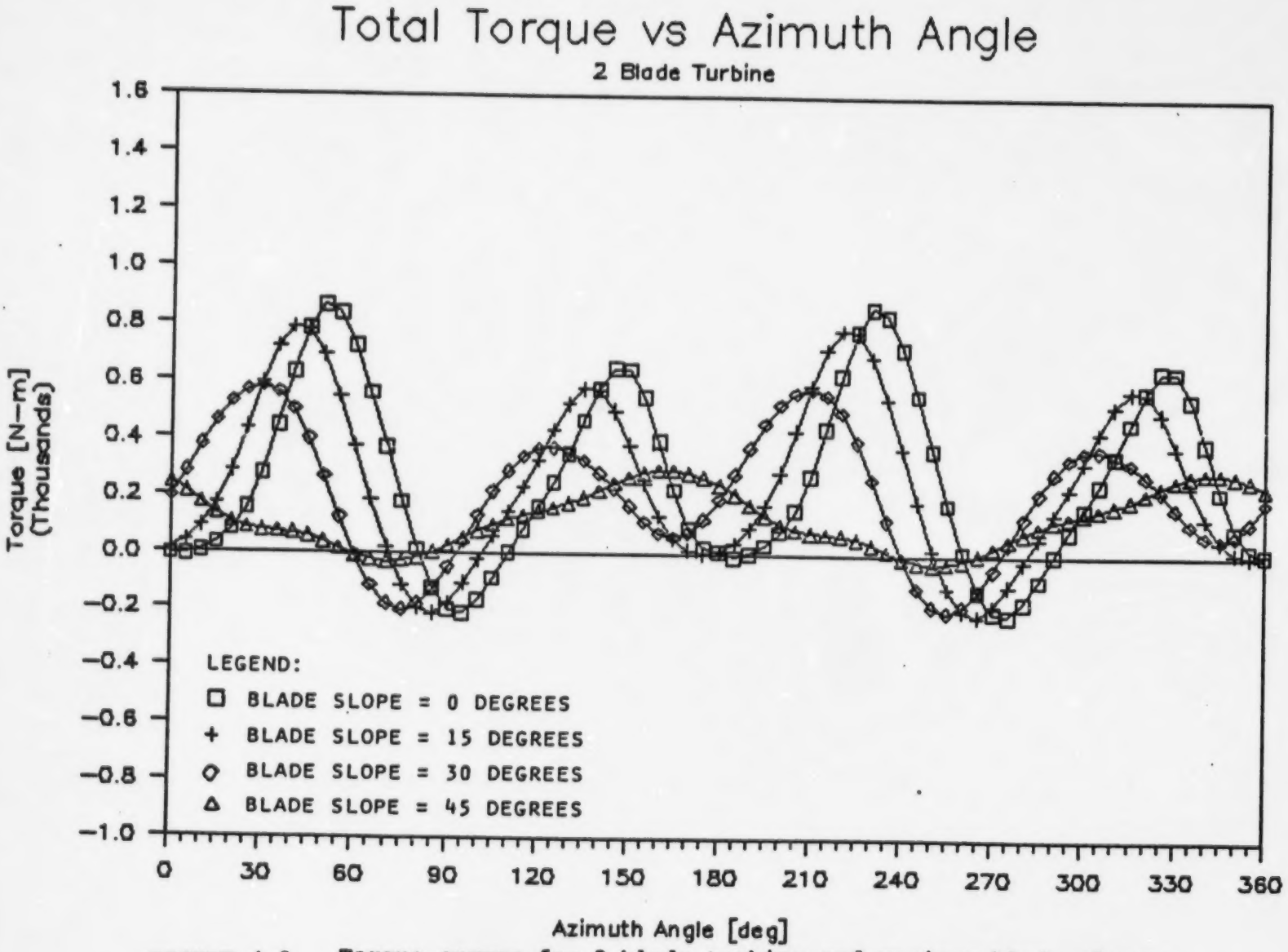


FIGURE 4.9 Torque curves for 2 blade turbine and various blade slopes

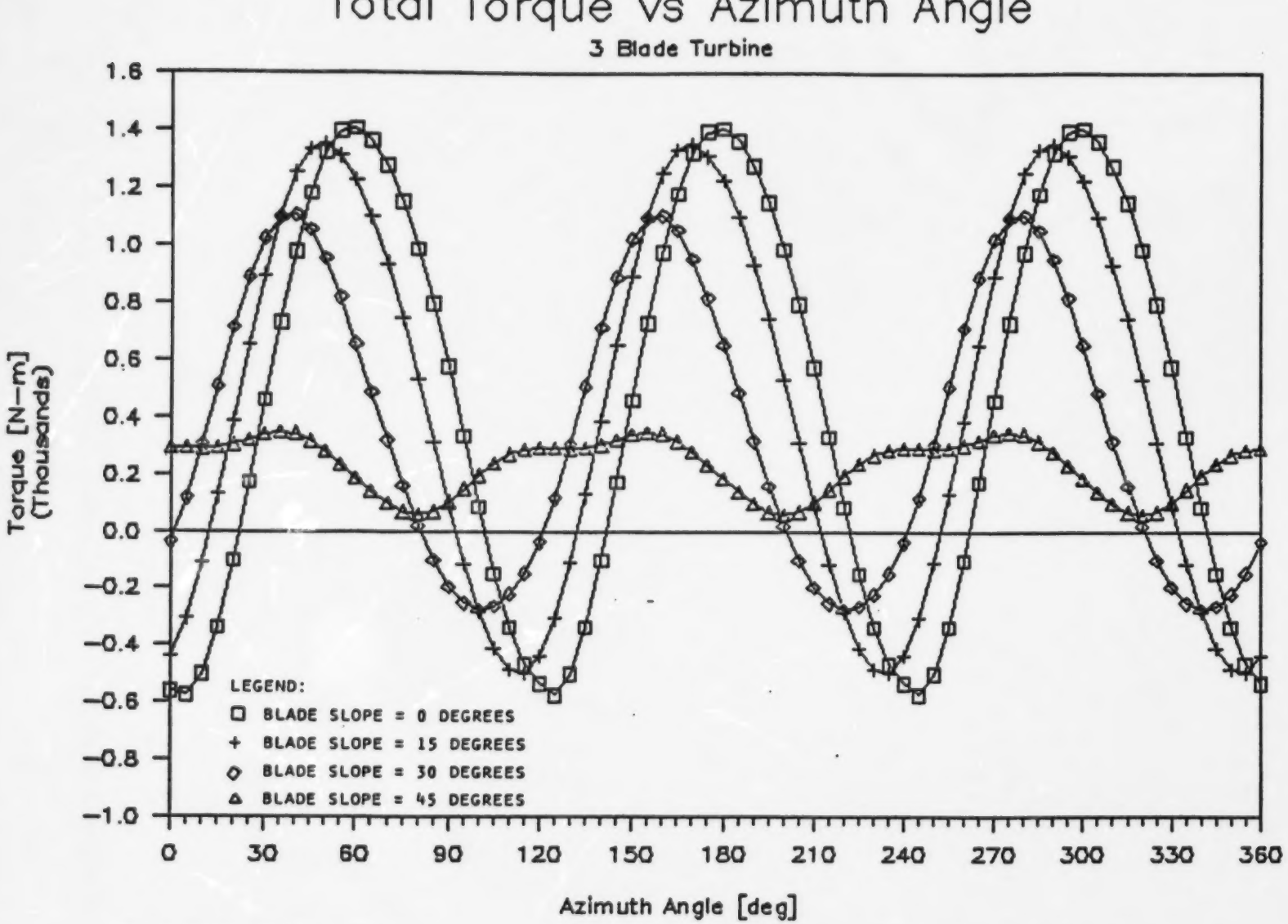


FIGURE 4.10 Torque curves for 3 blade turbine and various blade slopes

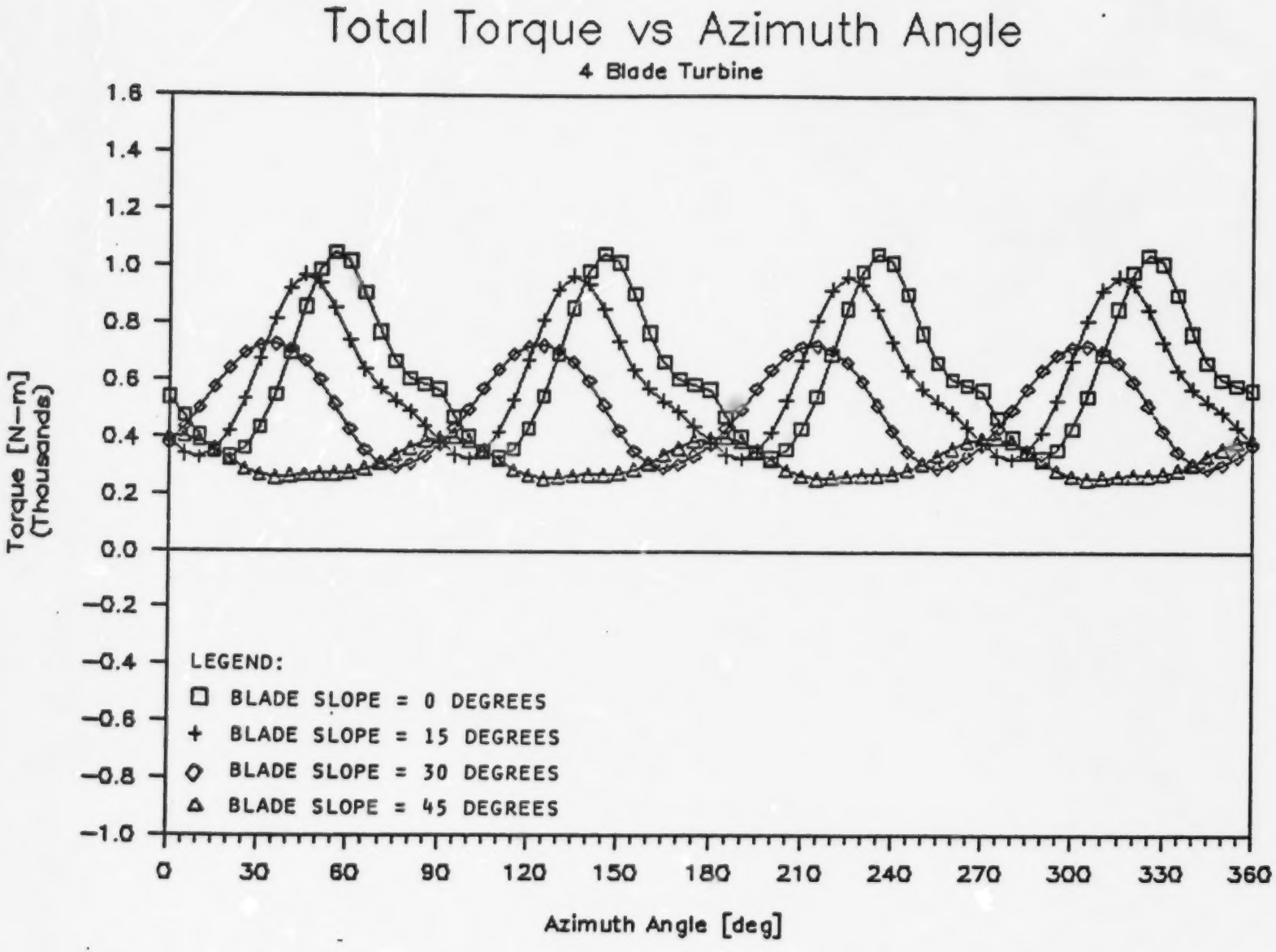


FIGURE 4.11 Torque curves for 4 blade turbine and various blade slopes

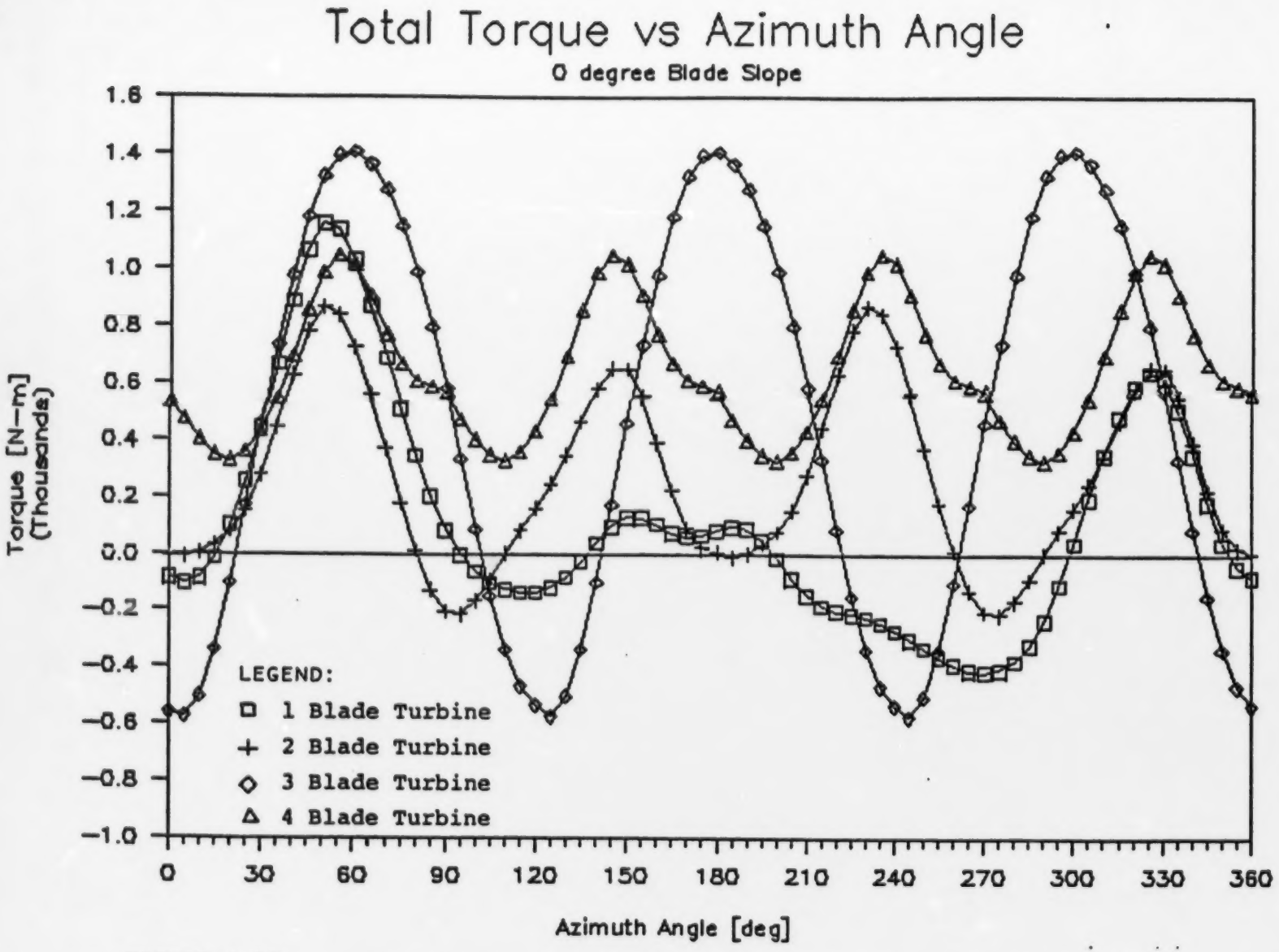


FIGURE 4.12 Torque curves for turbines having 1,2,3 and 4 blades with no blade slope.

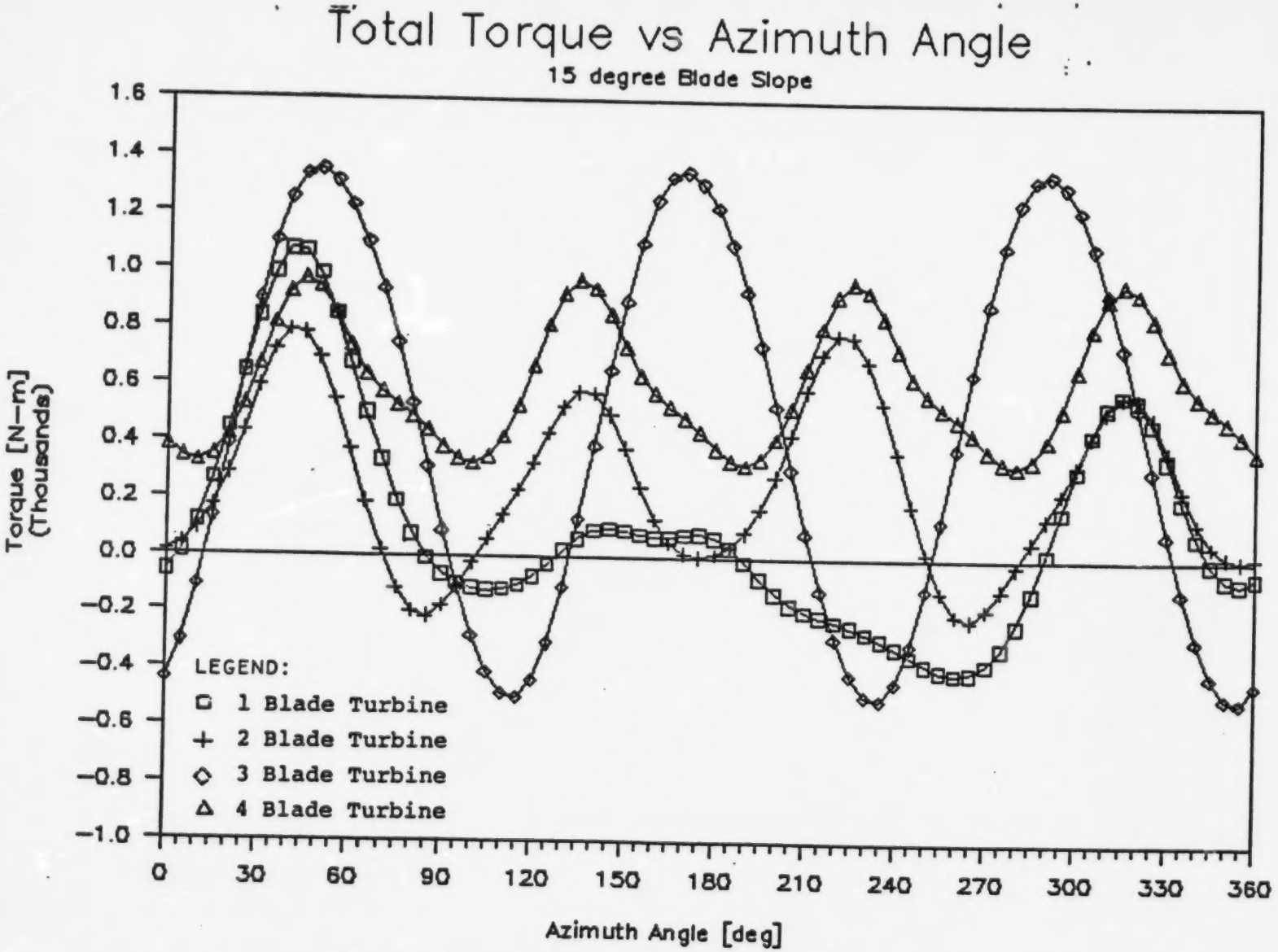


FIGURE 4.13 Torque curves for turbine having 1,2,3 and 4 blades with 15 degrees of blade slope

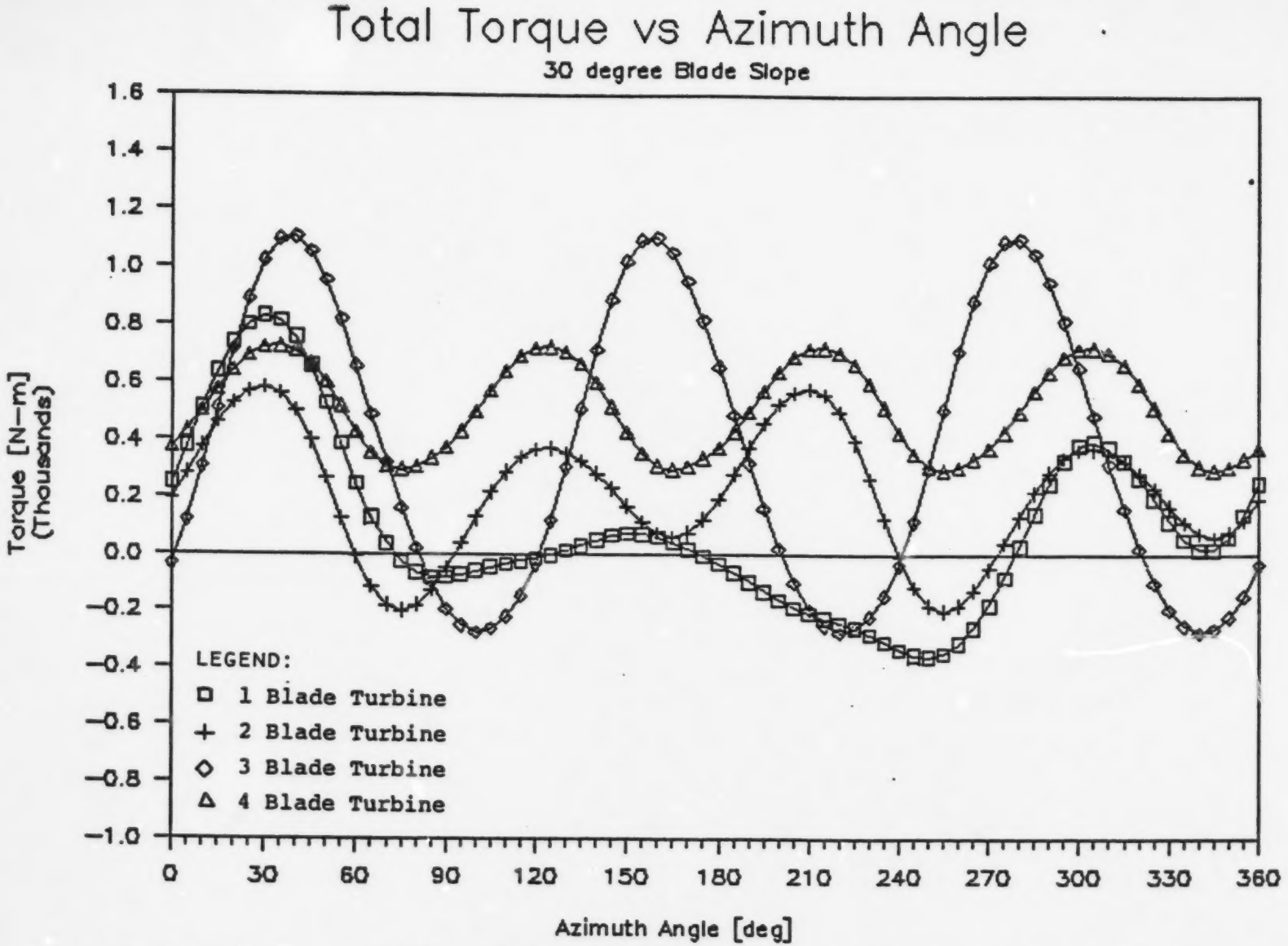


FIGURE 4.14 Torque curves for turbine having 1,2,3 and 4 blades with 30 degrees of blade slope.

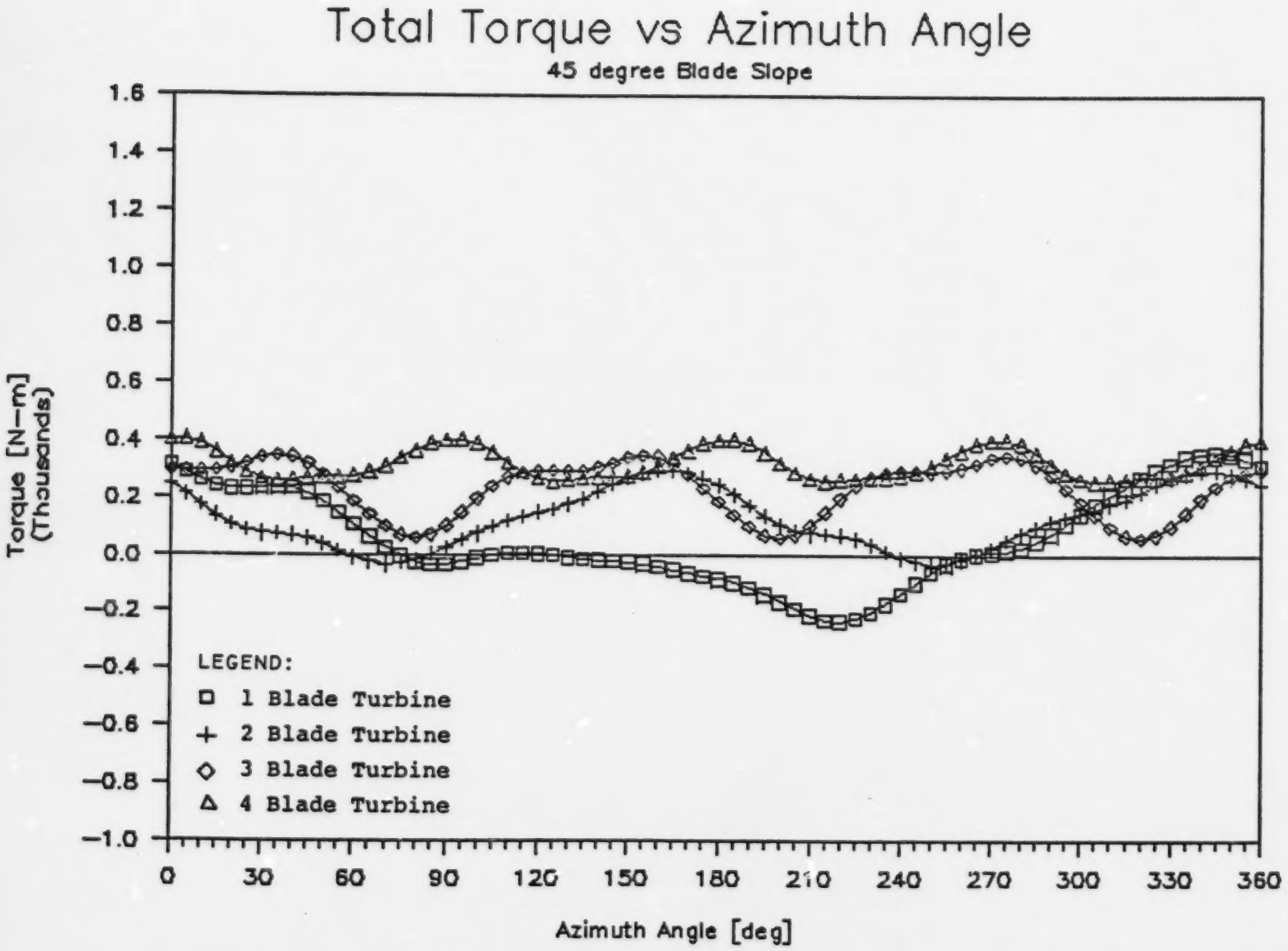
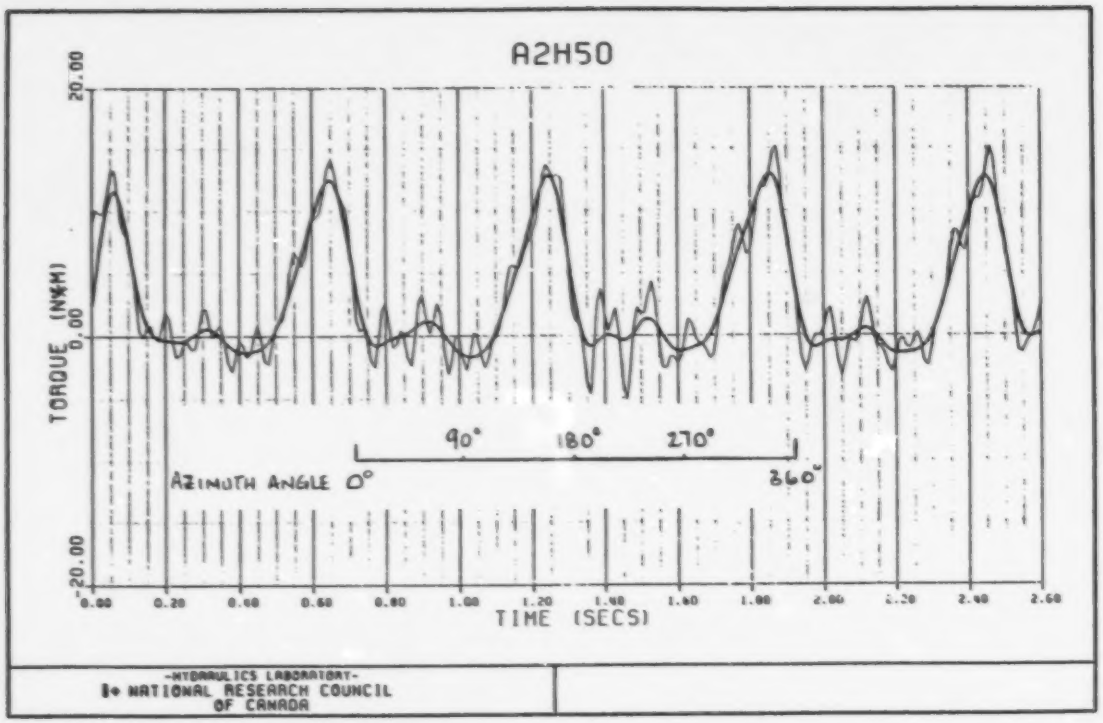


FIGURE 4.15 Torque curves for turbine having 1,2,3 and 4 blades with 45 degrees of blade slope.



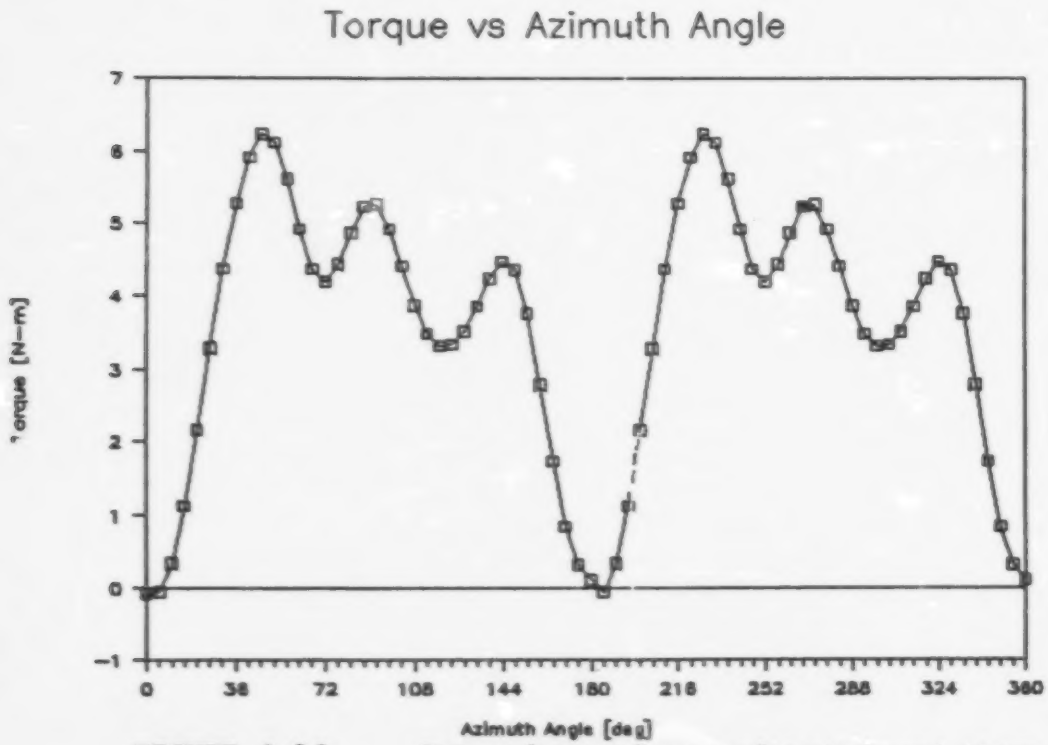
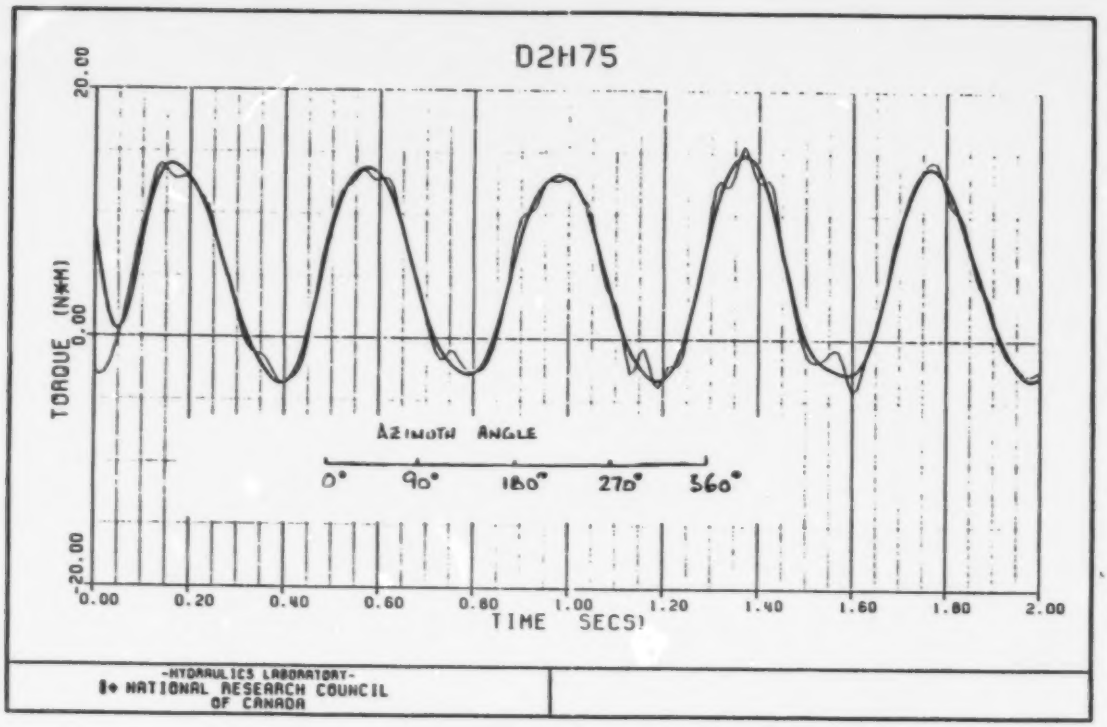


FIGURE 4.16 Comparison of experimental and theoretical results 2 blade turbine, 0^0 slope, 50 rpm



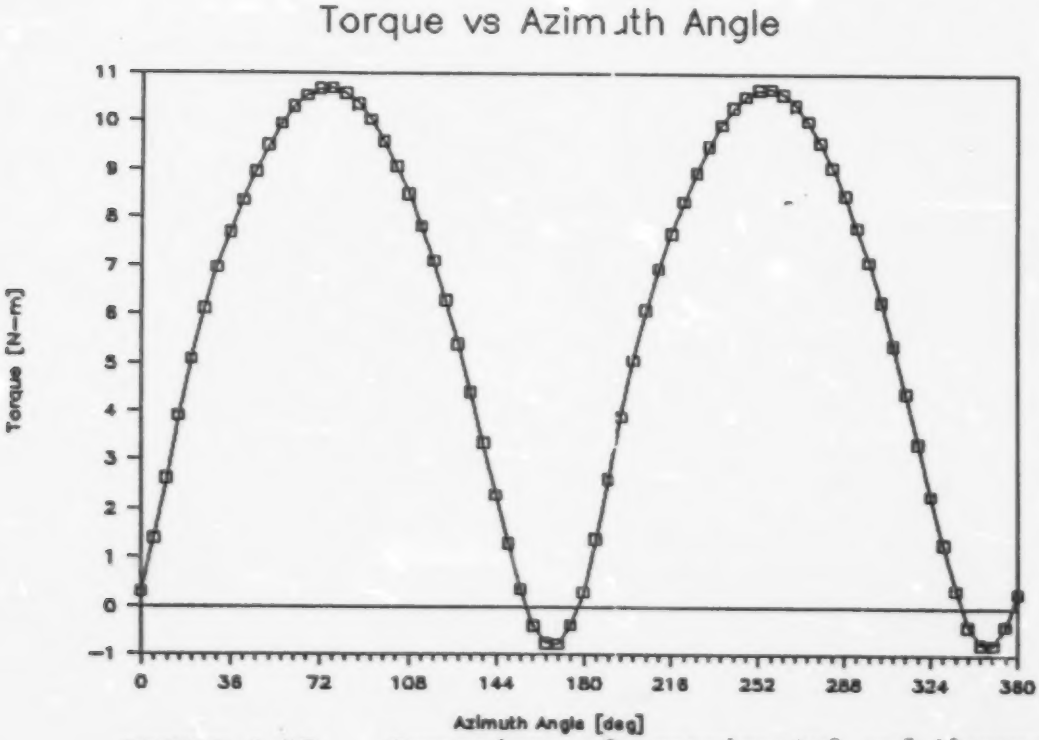
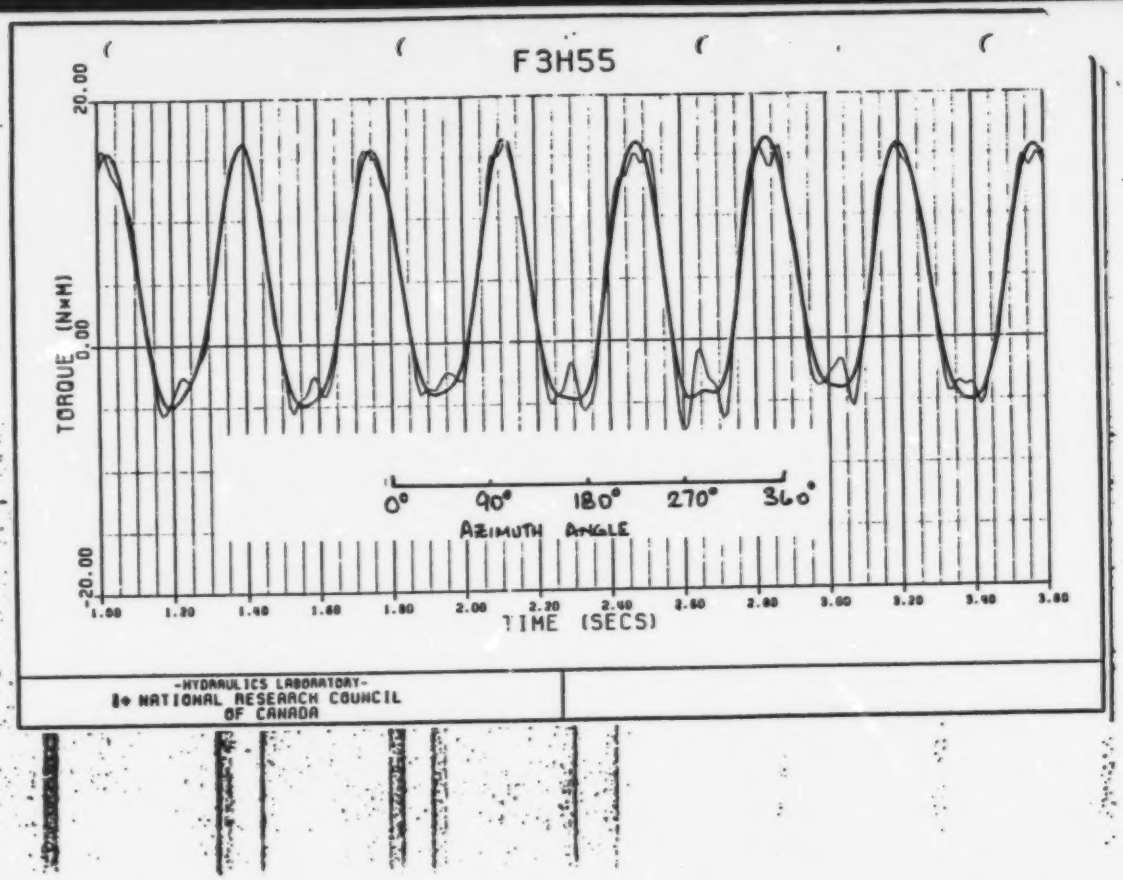


FIGURE 4.17 Comparison of experimental and theoretical results 2 blade turbine, 15.9° slope, 75 rpm



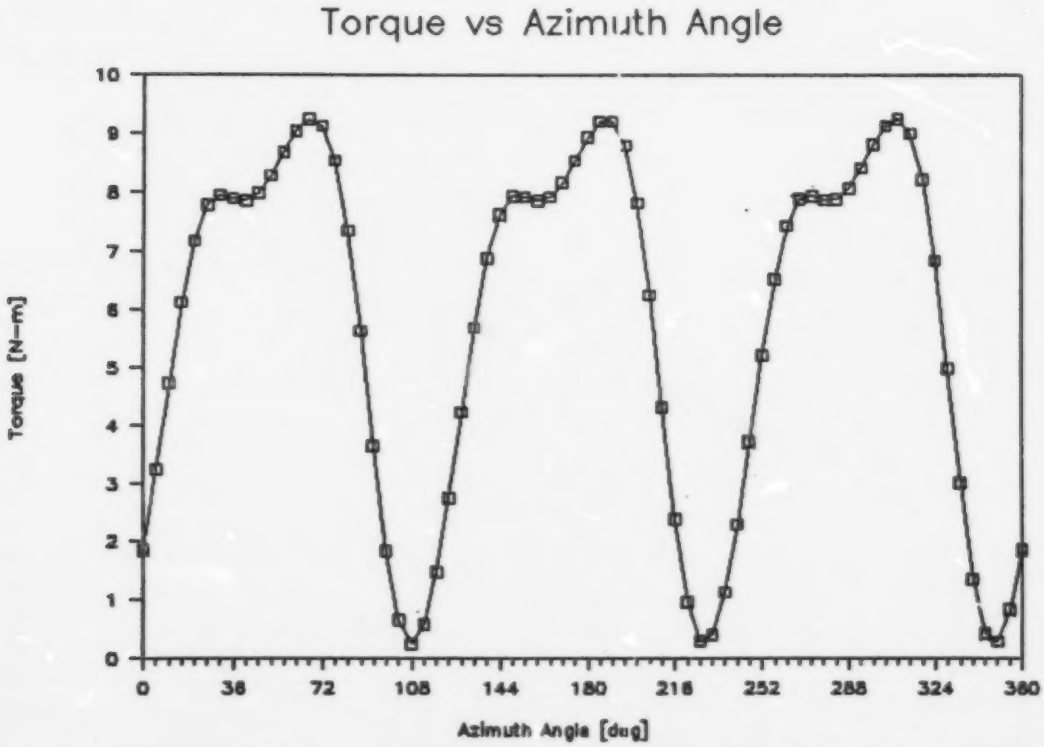
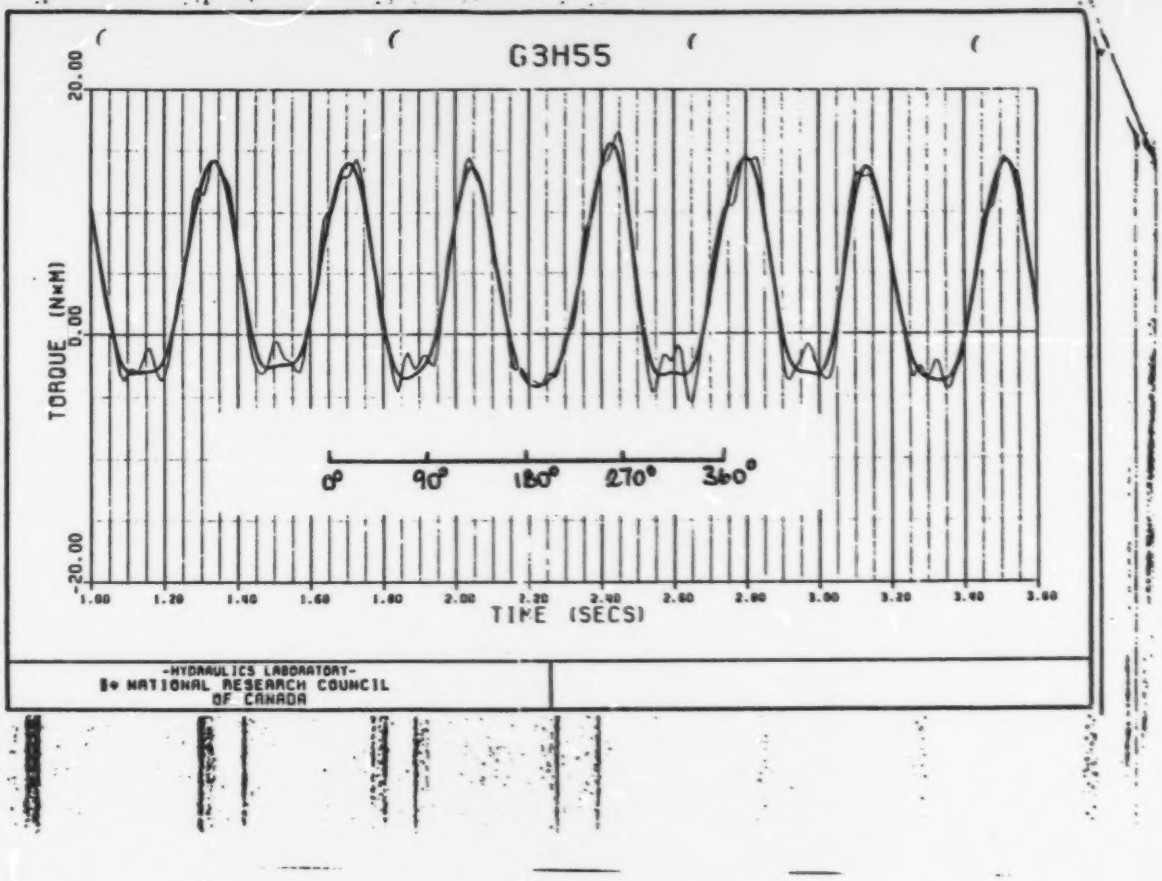


FIGURE 4.18 Comparison of experimental and theoretical results 3 blade turbine, 9.8^0 slope, 50 rpm



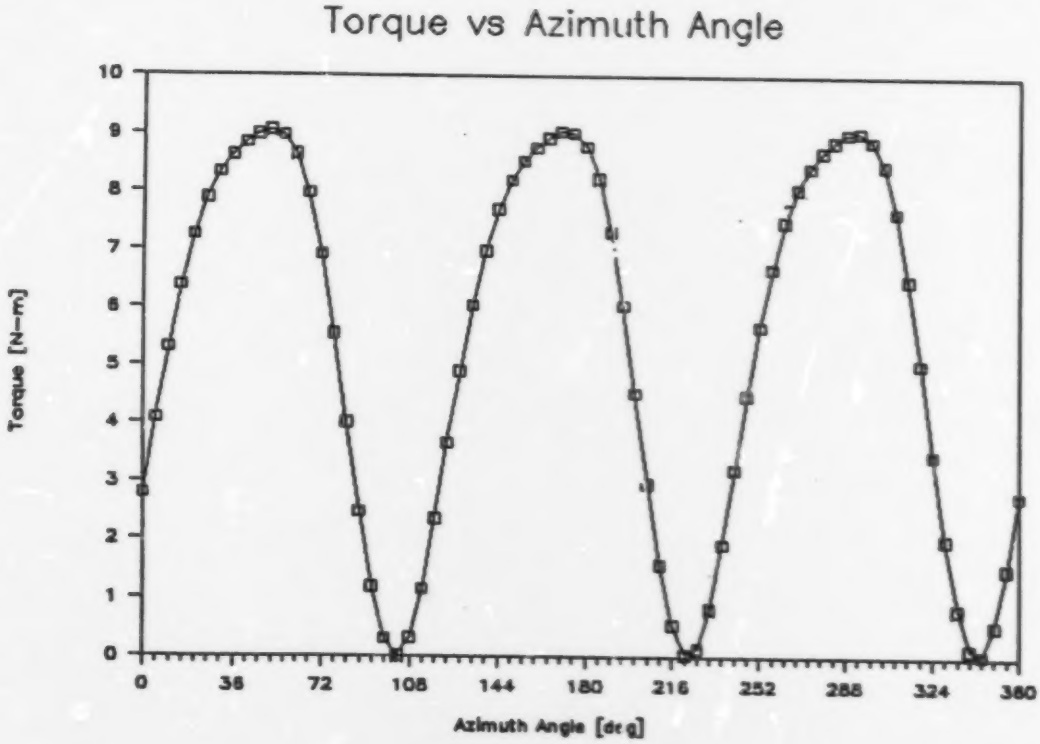
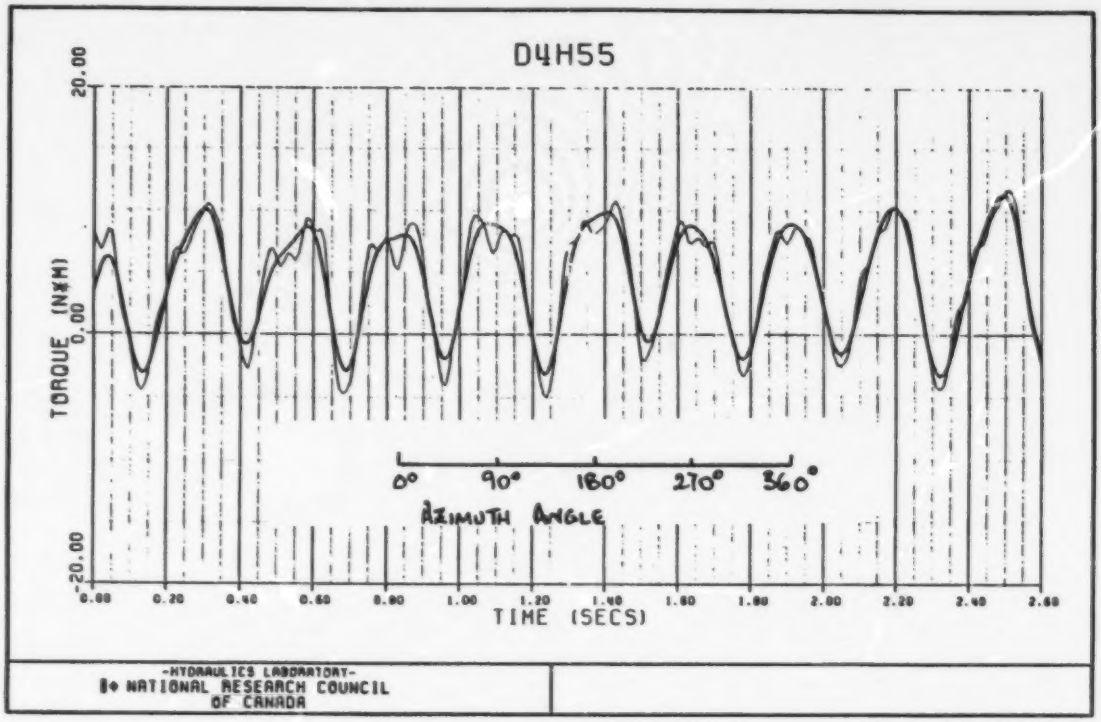


FIGURE 4.19 Comparison of experimental and theoretical results 3 blade turbine, 19.2° slope, 50 rpm



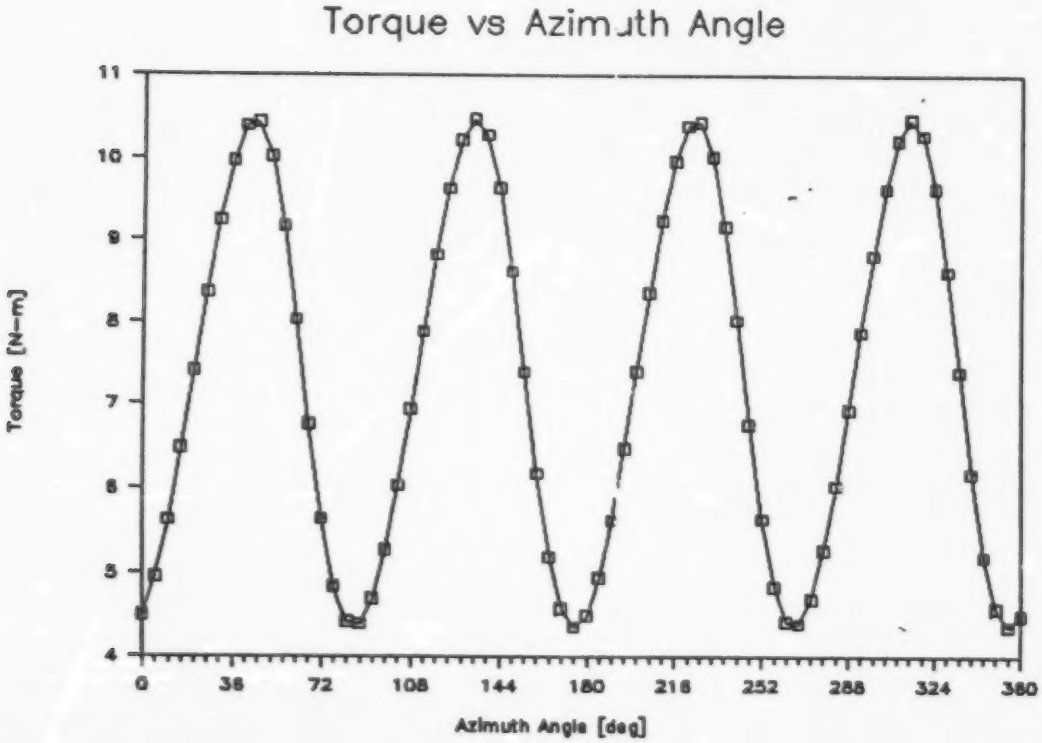


FIGURE 4.20 Comparisen of experimental and theoretical results 4 blade turbine, 15.9^0 slope, 50 rpm

5.0 CONCLUSIONS

5.1 Summary of results

It is clear from both experimental and theoretical results that, as expected, sloped blades result in torque smoothing but at the expense of power output. An attempt to quantify this (using the experimental results alone) resulted in Fig. 3.23 which shows a ratio of net average torque to the standard deviation of torque for different blade slopes. However, the degree of smoothing achieved in the test flume proved not to be as significant as predicted by the mathematical model, and the power reduction associated with sloped blades is somewhat greater than expected.

The mathematical model used does not appear to represent what is actually happening in practice, probably by failing to recognize any component of three dimensional flow and by not accounting for downstream wash from upstream blades. Computer prediction of the sloped blade turbine performance, while hopefully indicative of trends, cannot be confidently carried out until further computer program development is undertaken.

Reduced power from a sloped blade turbine was expected because of the smaller area intercepting flow, but the degree of reduction cannot be explained by this alone. We believe that a significant amount of the power deterioration is due to the axial lift component caused by the sloped blades (directing some of the water flow upwards), causing a non-uniform velocity gradient along the turbine axis.

A consequent higher velocity at one end of the turbine would exacerbate the torque pulse, perhaps explaining why this was not decreased to the extent predicted.

5.2 Influence of present work

When planning this present development work it was expected that the results for torque smoothing would be more conclusively positive. The scope of work therefore was initially aimed at optimizing the angle of blade slope (or azimuth angle between top and bottom of a blade) for torque pattern and power output. It is now apparent that the achievable torque smoothing is less than predicted from the two dimensional model, assuming uniform flow.

The sloped blade approach still shows some promise. The severity of torque pulse can be reduced and we believe that a helical bladed turbine with higher aspect ratio than that used in our tests (nominally 1:1) will show all round improved performance over "conventional" VAHT's.

The work has certainly furthered the understanding of this turbine type and indicates what subsequent development steps may be appropriate.

The question will be posed, and must be answered as to whether this type of design has a commercial future and is worthy of further development.

The demand for a viable low head hydropower device exists and Refs. 1 and 2 have examined and undertaken a cursory assessment of different low head turbine designs. No clear winners emerge and, in time, the market itself will decide those.

It is well to note that propeller-type turbines have a development history many times longer than the Darrieus-type and, in comparison, the latter (particularly the sloped bladed Darrieus) is poorly understood. Nevertheless the VAHT seems to have likely initial cost advantages over other types and, with an improved torque/power performance, should compete soundly in certain applications. On this basis further focussed development is clearly justified, offering as it does worldwide market opportunities for Canadian turbine manufacturers.

5.3 The market and its influence

This report makes no attempt to quantify the market for low head hydro systems, though recognizes that its size and location depends upon the operating cost-effectiveness. A turbine's mechanical power (from a rotating shaft) may find ultimate use in several applications such as: power generation, irrigation pumping, desalination by reverse osmosis, etc., and the power take-off required will be tailored to the specific application.

In addition, the siting of a turbine is important not only with regard to its load but also because of the characteristics of the energy available ot the turbine. The water flow can vary daily, seasonally, or reverse with the tidal cycle.

Clearly, the site conditions have a major influence on low head turbine economics, as with most diffuse energy concentration devices. Recognizing the sensitivity of such devices to their environment leads to an understanding of the difficulties faced in introducing renewable energy technologies. Unlike higher powered devices, they cannot compete using brute force. Lacking this they must be smarter which, while competing with relatively low cost fuels being used in "conventional" devices, is difficult to achieve now except in niche applications. This cross-over point moves as fuel costs change and technology improves. Nevertheless substantial niches exist now, both in Canada and

overseas for reliable and cost-effective devices.

5.4 Action arising

A further phase of development of sloped bladed Darrieus machines might include the following tasks:

1. experimenting with helical blades, using test-turbine developed under this present contract. This will quantify performance improvements of these blades (0, 10, 15, 25° blade slopes).

2. Experimenting with assimetric ducts to compensate for three dimensional flow effects.

3. Design and engineering of a roughly optimised pre-production prototype sloped bladed turbine. This would likely have 2:1 height to diameter ratio, with blade section and slope suggested by laboratory results and dimensions suitable for powered floating test platform.

4. Manufacture and testing, measuring power output by pumping water through calibrated nozzle.

In addition, other geometries can be devised to control the vertical component of flow whilst still spreading the torque generation for a blade over a larger azimuth angle.

Various turbine types have characteristics which make them suitable for different applications. Generally however, low head turbines are still at a stage of early and vigorous development and it is too early to draw definitive conclusions which may kill any generic turbine design. The Darrieus still shows promise and Darrieus type and Darrieus derivative turbines operate in a significantly different way from conventional axial flow turbines. Clearly, the development process is not yet complete and there are still indications that Darrieus type turbines will figure significantly in the growing market for low head power.

6.0 REFERENCES

1. Electricity from Water Currents. CEA Report No.: 357 G 449 Feb. 86

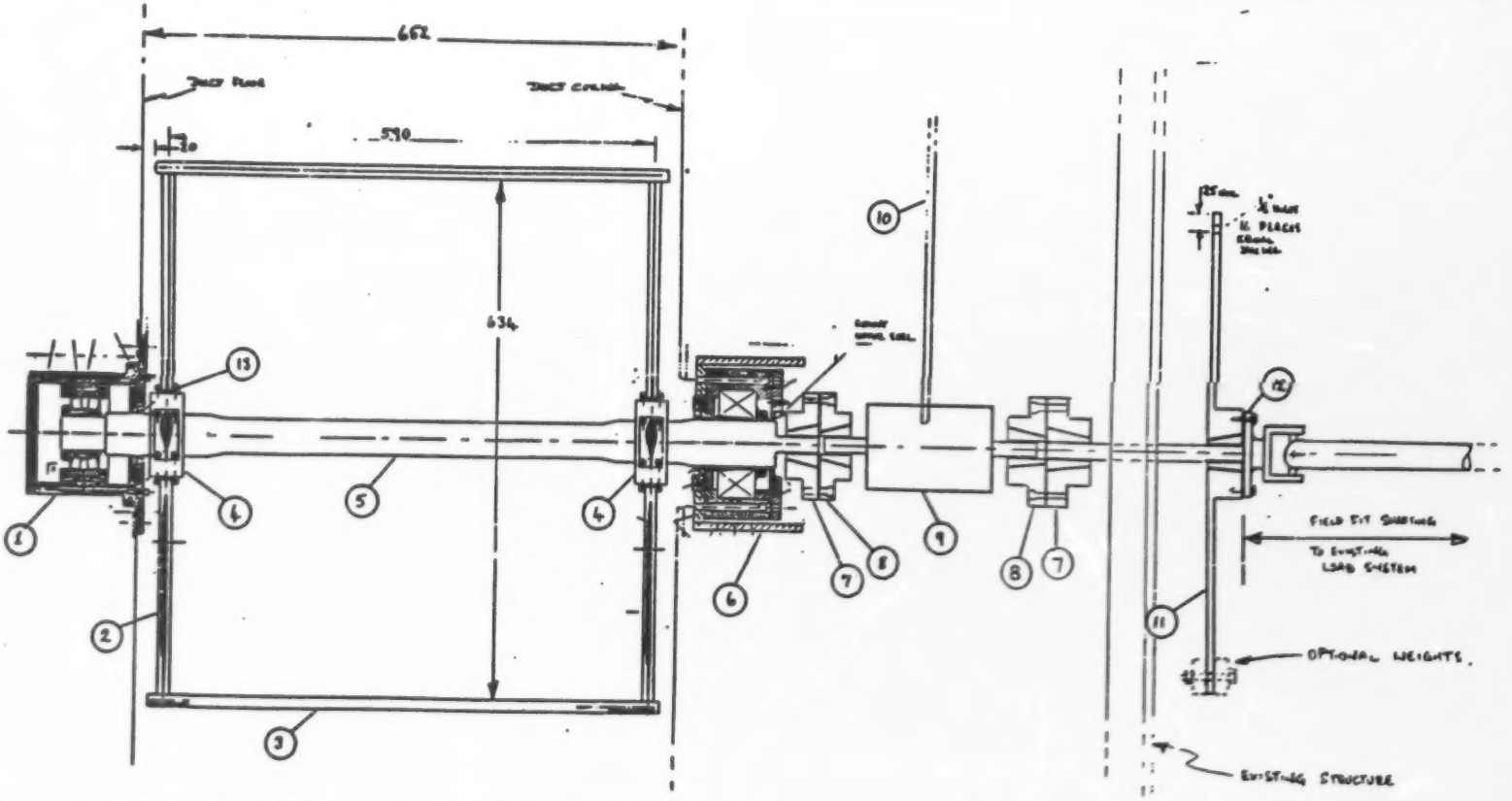
2. Ultra-Low Head Tidal Power: Technology and Site Assessment Highquest Engineering Inc. National Research Council contract no.: 19SR.31945-4-0027.

3. Test Report: Computer Program for Hydrodynamic Analysis of Vertical Axis Hydraulic Turbines. GasTOPS Ltd., Report GTL-TR-19-18.5. National Research Council contract no: 07SQ.31945-3-0039. March 1984.

4. Ultra Low Head Hydroelectric Power Generation Using Ducted Vertical Axis Water Turbines. Nova Energy Limited. July 1982. National Research Council contract no.: 06SX-31155-1-4502.

APPENDIX A

TEST BED TURBINE DESIGN DRAWINGS



13	22	SOCKET HEAD CAP SCREWS M8 X 15
12	1	ADAPTOR – SPROCKET HUB TO SHAFTING INSERT – FIELD FIT
11	1	PLUMMER – 600 $\frac{1}{2}$" PHR/SS TYPE TAPER-LOCK SPROCKET – DODGE # 100951 c/w 2517 BUSHING, $\frac{3}{4}$" BORE
10	1	ANTI-CENTRIFUGAL ARM FOR TORQUE TRANSDUCER – FIELD FIT
9	1	TORQUE TRANSDUCER
8	1	AS PER ITEM 7 INSERT ASSY DODGE # 007301
7	1	TAPER-LOCK HUB COUPLING, SIZE E16, $\frac{3}{4}$" BORE, FINISH ASSY DODGE # 007002
6	1	SHAFT BEARING ASSY EXISTING n/c
5	1	SHAFT HR DWG 1319-2
4	2	HUB – BLADE ARM – TAPER LOCK HR DWG 1319-3
3	4	BLADE (1 PR. DIFF) (SHOWN AT ZERO SPEED) HR DRAWING 1319-4
2	8	SHAFT ARM FINISH n/c HR DWG 1319-05
1	1	LOAD RESPONSE ASSY – EXISTING – NO CHANGE
ITEM QUAN		DESCRIPTION

DIMENSIONS IN mm

Highquest Engineering Inc.

DRAWN AFS	CHECKED JT	APPROVED RJF
DATE 8-11-96		

VAHT SLOPED BLADE TEST RIG

GENERAL ARRANGEMENT

1319-1

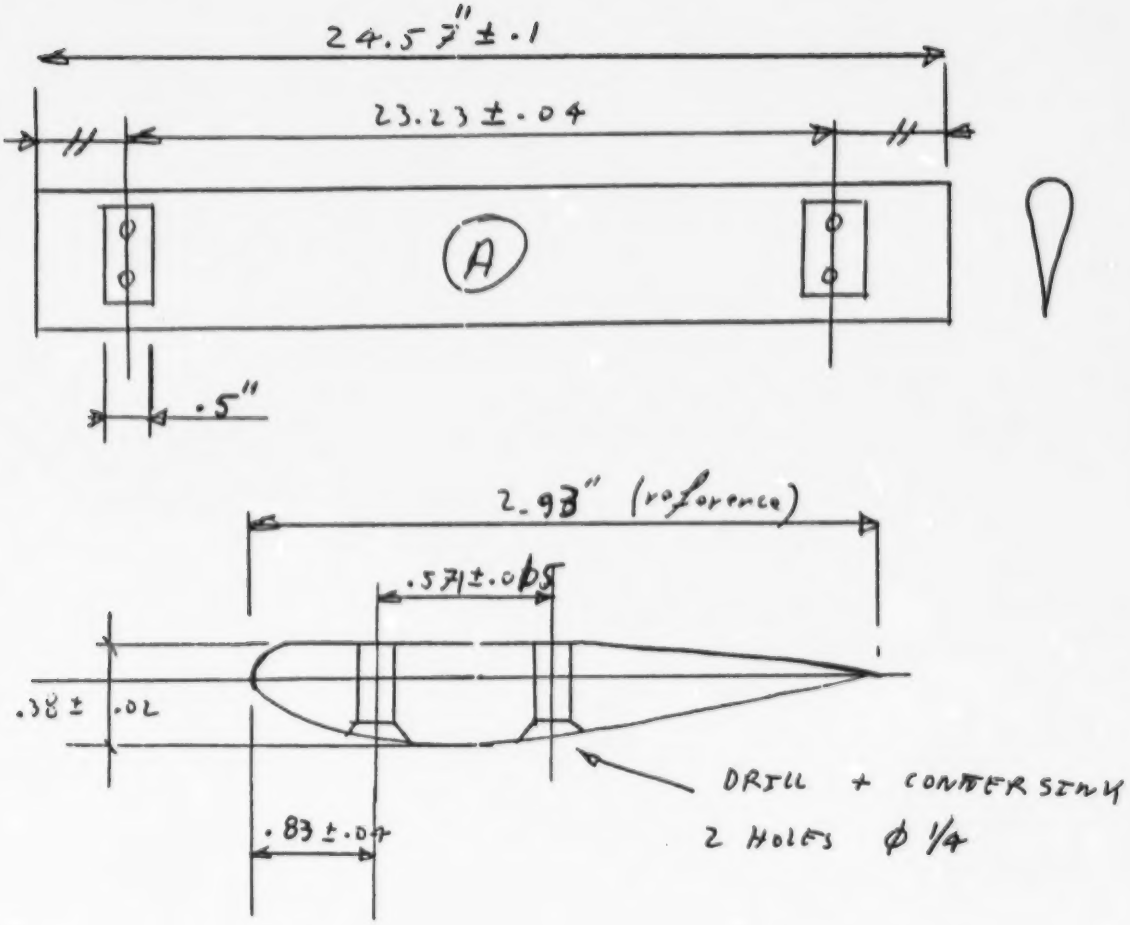
APPENDIX B

BLADE DIMENSIONS

CODE	α deg	θ deg	γ deg	β deg	CHORD cm	H cm	ROTOR HEIGHT cm	AB cm	BLADE LENGTH cm	AREA cm^2	RATIO	α whole blade	BLADE TWIST A-B (deg)
A	0	0	0	90	7.44	50	62.4	50	62.4	4043.5	1	0	0
E	15	8.2	1.1	81.9	7.51	50	62.4	50.6	63.1	4021.8	0.995	15.85	15.1
D	30	15.9	4.2	74.7	7.71	50	62.4	51.3	64.9	3957.4	0.979	31.64	31.1
B	50	24.9	11.1	67.6	8.05	50	62.4	55.1	68.8	3808.2	0.942	52.5	54.4
C	80	35.2	24.4	63.8	8.29	50	62.4	72.2	76.4	3466	0.857	83.18	91.5
A	0	0	0	0	7.44	49	62.4	49	62.4	4043.5	1	0	0
F	15	9.8	1.3	80.3	7.55	48.8	62.1	49.5	63.1	4009.1	0.991	19.1	12.6
G	30	19.2	5.2	71.5	7.85	48.1	61.3	51	64.9	3907.2	0.966	37.7	25.9
H	41.4	25.1	9.4	66.7	8.1	49	62.3	54.1	68.8	3918.5	0.969	51.3	45.3
I	65.1	35.5	21	60.7	8.53	49	62.2	60.2	76.4	3740.4	0.925	78.1	76.2

$$\text{RATIO} = \frac{\text{AREA}}{\text{NOM. RADIUS} \times 2 \times \text{NOM. HEIGHT}} = \frac{\text{AREA}}{62.4 \times 2 \times 32.4}$$

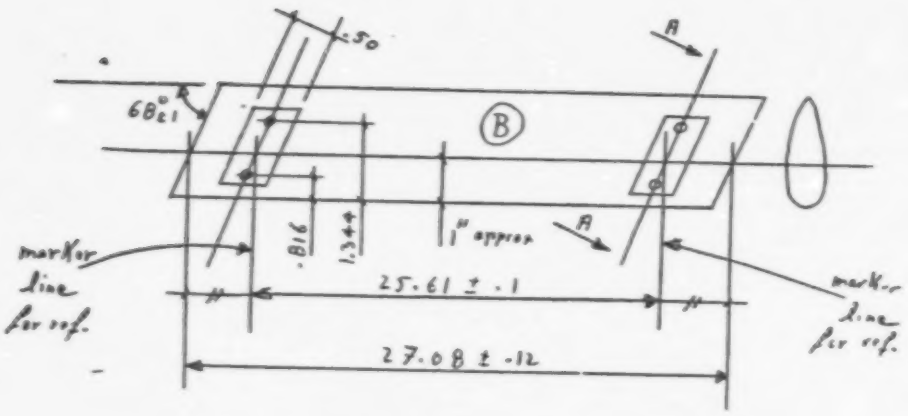
(See Fig. 2.1 for defn of angles)


24.5 7/8" ± .1
23.23 ± .04
A
.5"
2.98" (reference)
.571 ± .005
.38 ± .02
.83 ± .04
DRILL + CONTER SINK
2 HOLES ϕ 1/4

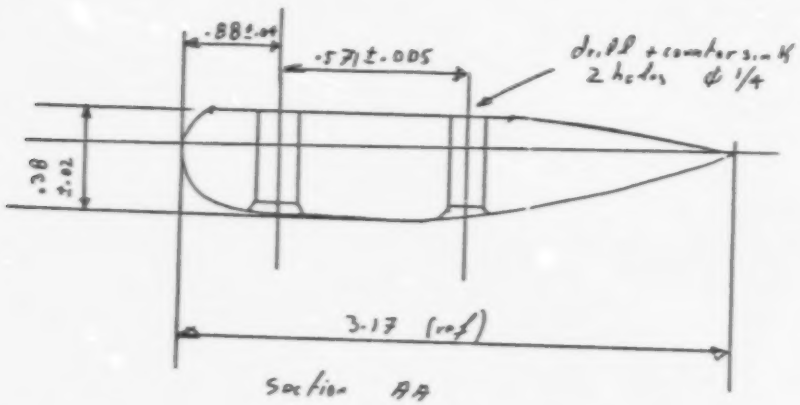
4 blades

4 blades of each type

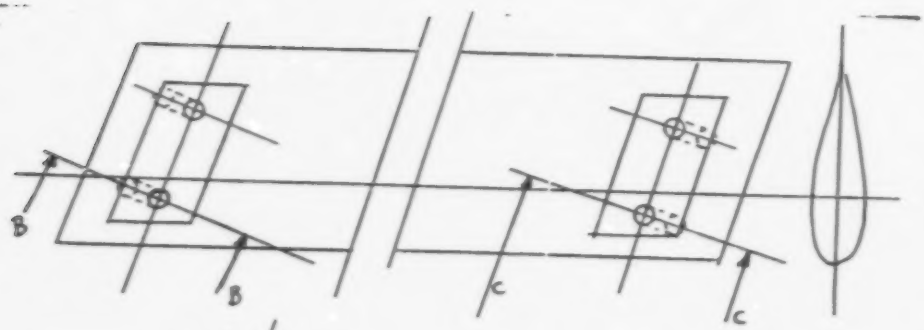
dim in inches (!!!)

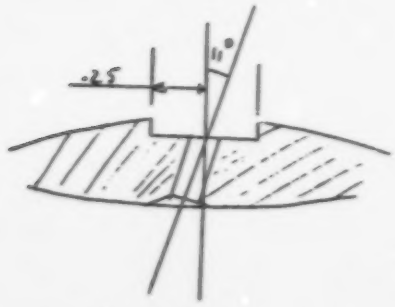


Blade is twisted only between the 2 marker lines.



Section AA





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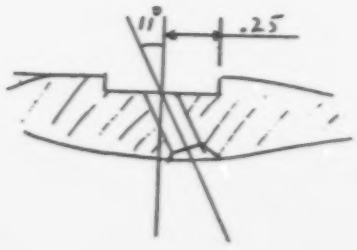
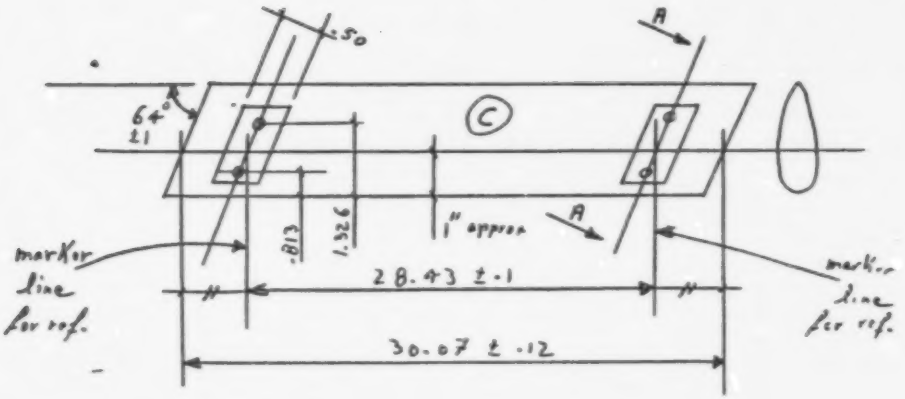


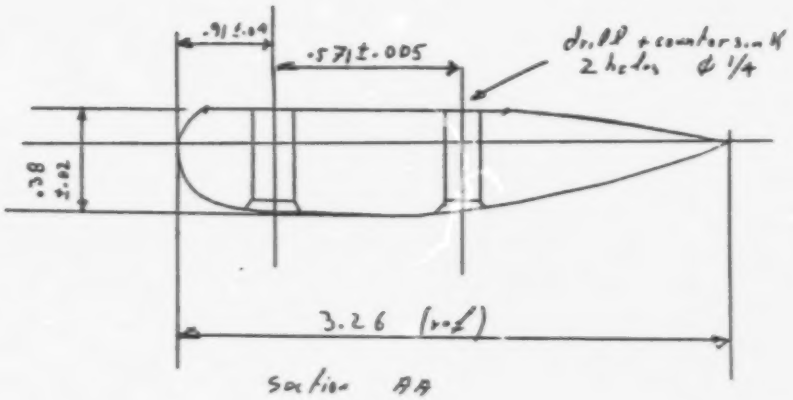
Section CC

4 blades of each type

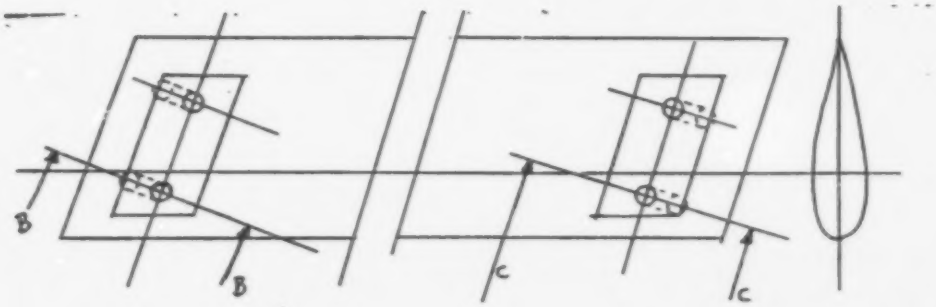
dim in inches (!!!)

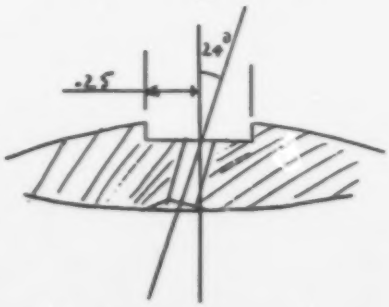


Blade is twisted only between the 2 marker lines -

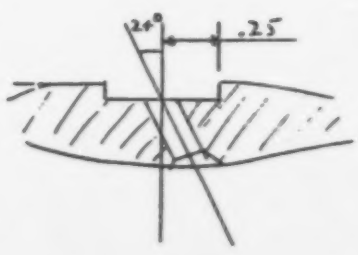


Section AA





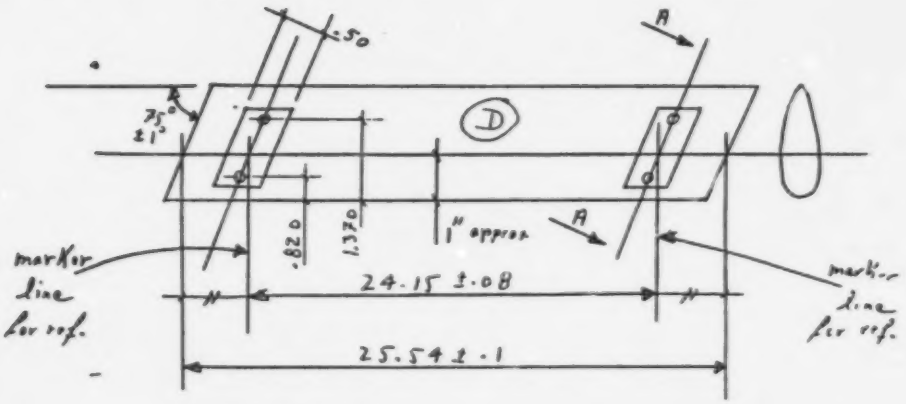
Section BB



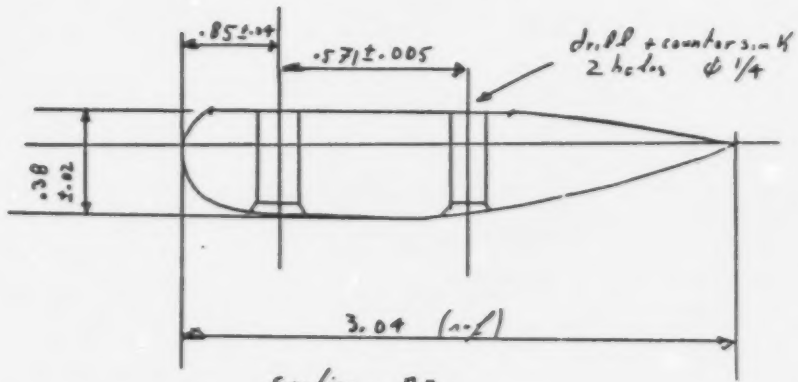
Section CC

4 blades of such type

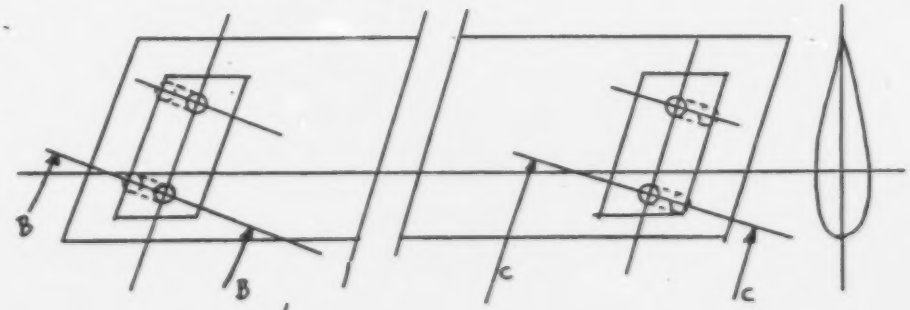
dim in inches (!!!)

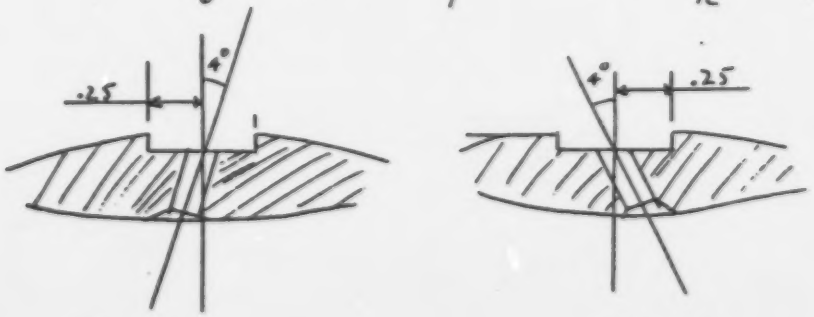


Blade is twisted only between the 2 marker lines -



Section AA



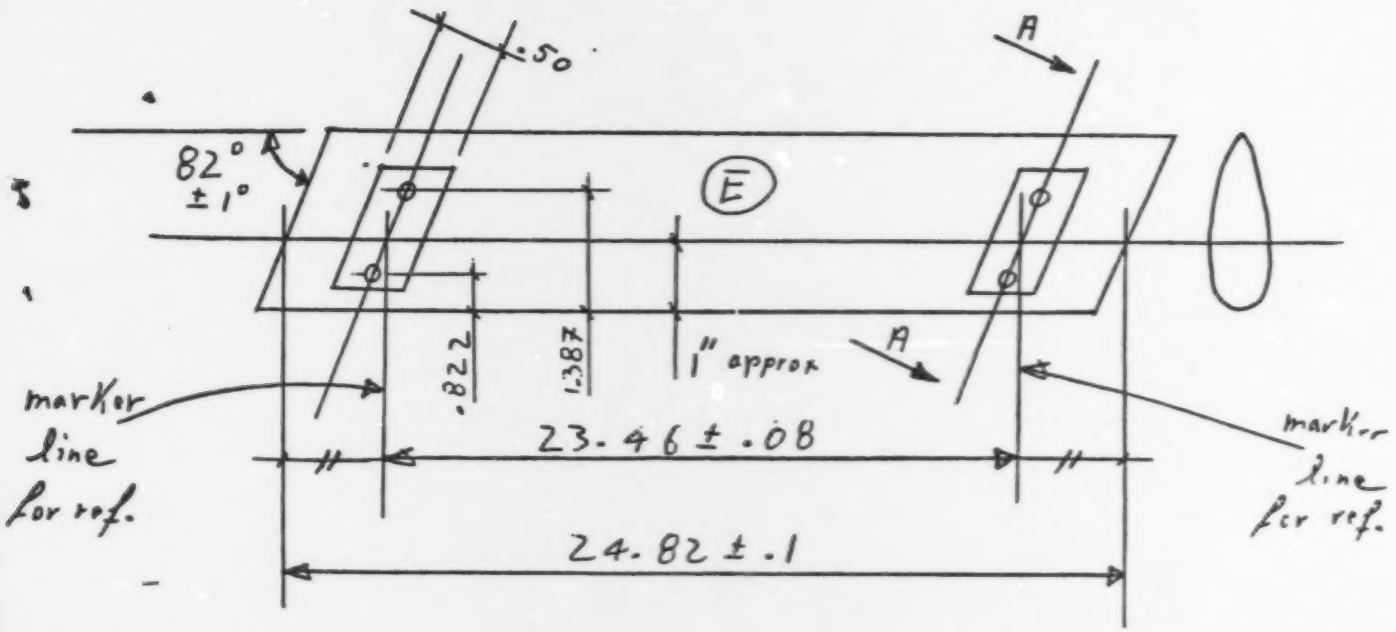


Section BB

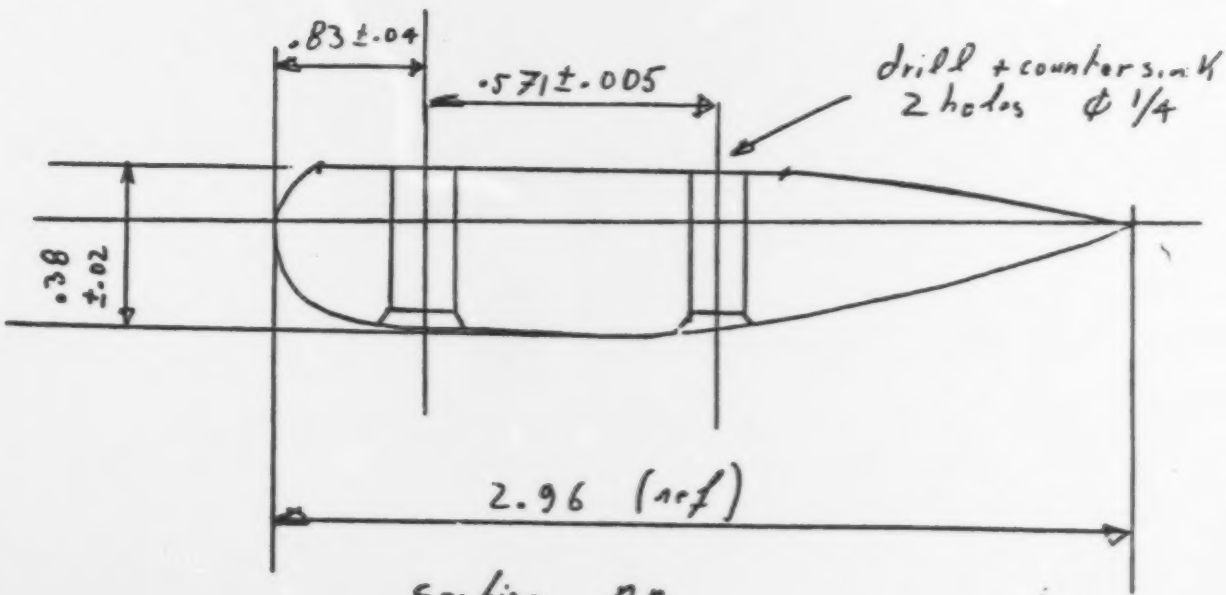
Section CC

4 blades of each type

dim in inches (!!!)



Blade is twisted only between the 2 marker lines -



Section AA